Proceedings of the International Conference on

LIQUEFIED NATURAL GAS

LONDON March 1969

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British Cryogenics Council
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Foreword.

This Conference comprised one half of an overall meeting on the theme "Cryogenics in the Fuel and Power Technology" organised by the British Cryogenics Council in London in March 1969, in conjunction with the 1st Commission of the I I R. The planning and detailed work of the LNG Conference was carried out by a Committee of the B C C chaired by Dr. Martin Ruhemann, and much of the editorial work was shouldered by Dr. Derek Ward.

Secretarial assistance was provided by the Institution of Mechanical Engineers whilst premises for the meeting were granted by the Institution of Electrical Engineers.

It is one of the strengths of the British Cryogenics Council that it can turn to many organisations, and many able low temperature scientists, for assistance, and their help is gratefully acknowledged.

G.G. Haselden
Chairman of British Cryogenics Council
Vice President 1st Commission I I R .

WORLDWIDE LNG MARKETS

A. R. Khan, Technical Director, Gas Developments Corporation
T. J. Joyce, Associate Director, Institute of Gas Technology

Introduction

The past 20 years have seen a tremendous upsurge in the consumption of all forms of energy throughout the world. Major factors in this upsurge are the formation of economic blocs, technological developments, and the greater energy requirements of increasingly industrialized countries.

An idea of the growth of world energy consumption can be gotten from Figure 1. Until 1952 (except for the United States), almost 80% of this energy consumption was met by solid fuels, with natural gas accounting for less than 1%. Even as late as 1962, nearly 41% of Europe's energy requirements were met by solid fuels, while liquid fuels and electricity accounted for another 47%. Natural gas supplied only 2.5% of all energy consumed for that year, with manufactured gas providing the balance. The development of the economic blocs — the European Economic Community (EEC) and the European Free Trade Association (EFTA), and their Communist counterparts — has resulted in a huge demand for energy in the highly industrialized countries of Europe. These countries, together with Japan, constitute the large energy-hungry market that will grow rapidly in the next decade. What share of this energy market will belong to natural gas? As can be seen from Figure 1, natural gas increased from 4% of total energy consumption in 1930 to 12% in 1952 and to 16% in 1962, and is projected to reach 20% by 1970.

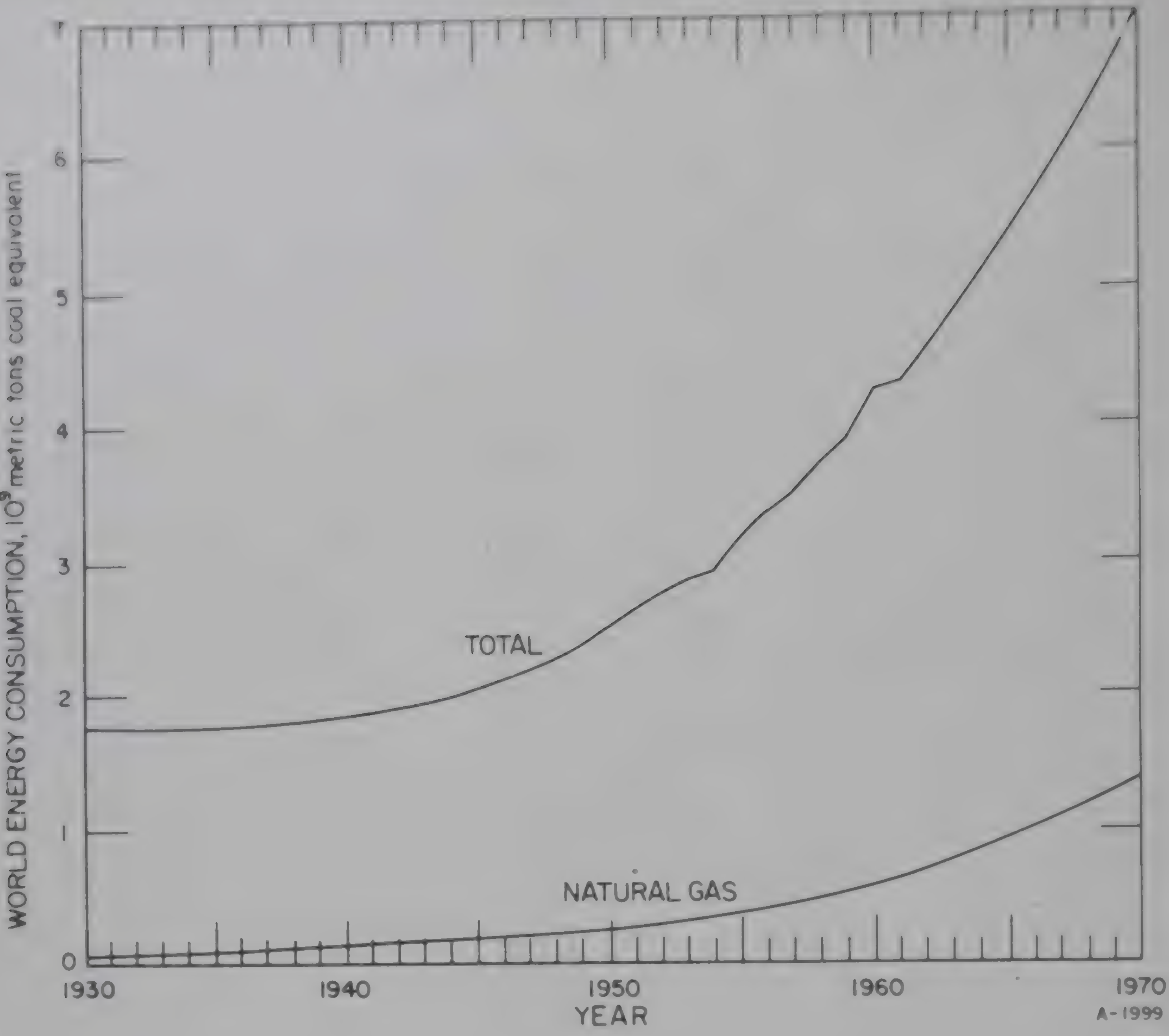


Figure 1. GROWTH OF WORLD ENERGY CONSUMPTION

The conventional method of transporting energy in the form of natural gas has been by pipeline. However, the increasing role occupied by natural gas in the world energy picture has resulted in the development of techniques for moving this energy from major supply sources to gas-deficient areas in the form of liquefied natural gas (LNG).

To determine the role that LNG can play in worldwide markets, it is necessary to examine the availability of natural gas, the location of reserves, consumption patterns, and the overall supply-demand picture that the future portends.

Natural Gas — World Supply and Demand

Gas Supply

The principal consumption of natural gas is in North America, Western Europe, and the European Communist countries. However, the distribution of natural gas reserves in the world is widespread. Major gas reserves occur in the United States, the Soviet Union, North Africa, the Middle East, and parts of Western Europe. The regional distribution of natural gas reserves is presented in Table 1.

Table 1. DISTRIBUTION OF WORLD NATURAL GAS RESERVES
Source: World Oil 167, (1968) August 15

	Estimated Reserves at End of 1967, 10^{12} CF
North and Central America	351.94
South America	48.93
Western Europe	118.36
North Africa	112.52
Central and Southern Africa	3.47
Middle East	228.24
Far East	33.01
Oceania	10.45
Communist Countries	164.97
World Total	1071.89

The largest reserves in any single country are in the United States: 292.91 trillion CF. Of the 118.36 trillion CF cited for Western Europe, 68 trillion CF are located in The Netherlands and 25 trillion CF in the United Kingdom. The Soviet Union accounts for about 150 trillion CF.

Excluding these gas reserves in developed countries, major reserves are located mostly in underdeveloped or developing nations in the Middle East, Far East, and North Africa. The exploitation and consumption of these reserves will largely be determined by the availability of markets and the economics of supply overseas as LNG at competitive prices. The possible sources of LNG supply are presented in Table 2. Since most gas production has been associated with oil production in the past, estimates of gas reserves are usually conservative and do not include accurate estimates of gas fields that have been shut in. This is particularly true for fields in the Middle East where the primary concern has been oil production and gas has been of secondary value.

Gas Demand

Over 70% of the world's consumption of natural gas is in the United States. The balance of the demand for natural gas energy is in Canada, Western Europe, and the Eastern European Communist countries. In the Eastern Hemisphere, Japan has a large consumption of natural gas and this demand is increasing rapidly. Projections of future gas consumption indicate that within the next 10 years certain countries in Europe will be importing gas either as pipeline gas or LNG. Some of these import schemes are already in progress. Table 3 presents estimated imports of gas for selected countries by 1978.

Table 2. POTENTIAL SOURCES FOR LNG SUPPLY
Source: World Oil 167, (1968) August 15

	Country	Estimated Reserves, 10^{12} CF
Africa	Algeria	100
	Libya	12.5
	Nigeria	3.0
Middle East and Far East	Saudi Arabia	50.0
	Abu Dhabi and Neutral Zone	12.0
	Kuwait	35.0
	Qatar	7.0
	Iran	100.0
	Iraq	21.0
	Pakistan	20.5
	Indonesia	2.0
Western Hemisphere	Canada	45.7
	Chile	3.0
	Colombia	4.0
	Venezuela	27.8
	Mexico	11.8

Table 3. GAS IMPORTS FOR SELECTED COUNTRIES – 1978
Source: Institute of Gas Technology

Country	Gas Imports, 10^9 CF/yr
Austria	70
Belgium	177
France	400
Germany, West	600+
Italy	460
Portugal	20
Spain	175
Czechoslovakia	75
Yugoslavia	30
Japan	300
U. S. A.	1855

These estimated figures indicate that natural gas consumption will increase sharply in the EEC countries, Southern Europe, and Japan. Most of this demand will, of course, be met by large transmission lines that will bring gas from the fields in The Netherlands and the Soviet Union. However, the areas bordering the Mediterranean Sea will also have the option of purchasing imported LNG if the gas trunklines being constructed in Europe do not extend sufficiently far south and if the economics of LNG imports are more attractive than pipeline gas. For Japan, all of the imported gas will be LNG unless the proposal to transport Soviet gas by pipeline from Sakhalin to northern Hokkaido becomes a reality.

However, the development of LNG projects is not solely dependent on market demands. The economics of energy supply and demand dictate that the delivered price of imported LNG must be less than, or competitive with, other fuel on a Btu basis, making suitable allowances, of course, for such special advantages as pollution abatement and so forth. Also, since the investment required is of a considerable magnitude, assurances and guarantees of the source of gas supply are necessary, particularly if the gas comes from developing countries that are politically unstable.

LNG Import-Export Projects

The first LNG scheme to export gas from Algeria to the United Kingdom and France started operation in 1964. Since then, a number of additional projects have been contracted for.

Table 4 presents a list of LNG import schemes that are either in operation or soon to be in operation. In addition, some of the LNG import projects presently under study and consideration are —

Venezuela-Algeria to East Coast, U. S. A.

Venezuela-Algeria to Canada

Algeria to Spain

Algeria to Italy, Yugoslavia and Austria

U. S. S. R. to Japan

Abu Dhabi to Japan

Table 4. OVERSEAS LNG PROJECTS

No.	Company	Country of Export	Country of Import	Total Imports, 10^6 CF/day	Project Status
1	Conch Methane Gas Council	Algeria	United Kingdom	100	Completed
	Gaz de France		France	50	
2	Esso International ENI	Libya	Italy	235	Completed
	Catalana de Gas		Spain	110	
3	Union-Marathon Tokyo Gas	U. S. (Alaska)	Japan	100	Engineering
4	Gaz de France Sonatrach	Algeria	France	450	Engineering
5	Royal Dutch/Shell Teikoku Oil Co.	Brunei	Japan	200	Planning

Figure 2 shows the location of major gas reserves that could serve as sources of LNG and the routes from these reserves to potential markets.

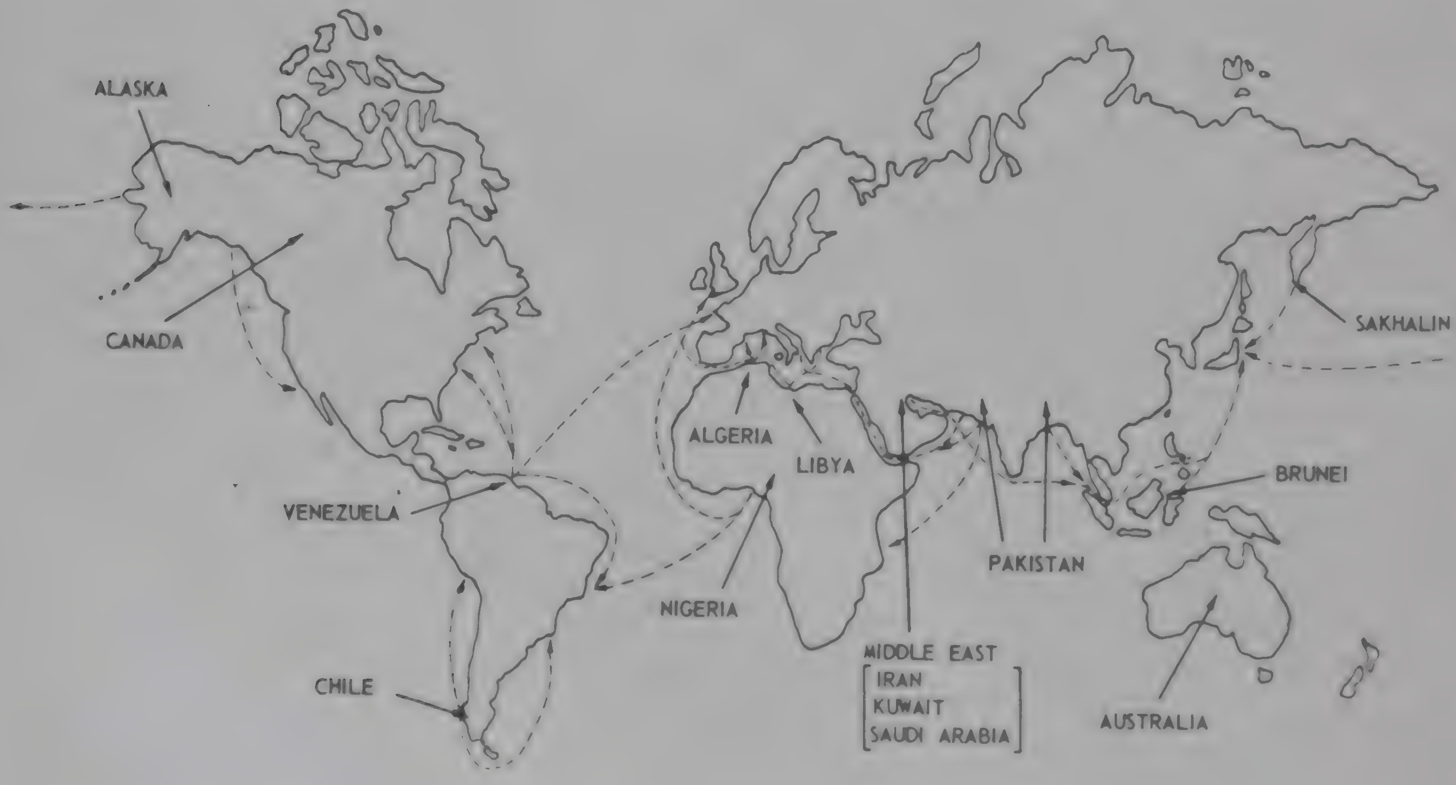

ALASKA
CANADA
VENEZUELA
CHILE
ALGERIA
LIBYA
NIGERIA
MIDDLE EAST
IRAN
KUWAIT
SAUDI ARABIA
PAKISTAN
SAKHALIN
BRUNEI
AUSTRALIA
A-1071435

Figure 2. POTENTIAL LNG SOURCES AND MARKET ROUTES

After determining the availability of gas at these sources and the presence of potential markets, we next need to determine whether such projects are economically feasible.

Economics of LNG Import Projects

An LNG import project is defined as a base-load gas supply scheme. These projects differ from the peakshaving LNG plants in the United States in that the size of these plants is larger, although storage requirements are not in the same ratio. Also, most of these projects require major investments in pipelines for moving gas from the fields to the liquefaction plant and also from the delivery port to a transmission grid. Investment costs in the exporting country include the construction of harbor and loading facilities, labor housing, site development, and a host of other costs normally not associated with a peakshaving LNG plant. The total investment required at the exporting end includes the above costs in addition to the investment for the liquefaction plant and the storage facilities.

The transportation of LNG by tankers requires a major investment for the ships. LNG tanker costs are higher than those for oil tankers of similar capacity, but the unit shipping cost for LNG decreases almost 25% when twice as much LNG per year is transported over the same distance. The effect of distance on shipping costs is fairly linear; hence, the landed cost of gas is more a function of the total annual volume and distance. With developments in LNG tanker design, tanker investment is expected to be reduced, but, for the present, large-volume movement seems to offer lower shipping costs.

Taking into account all these factors, it is possible to compute the unit cost for a total annual delivery over a particular distance providing the base cost of feed gas is known. Breakdown of this unit cost into components attributable to liquefaction, shipping, and raw feed gas costs indicates that the liquefaction plant (including storage and dock facilities) accounts for almost 50% of the delivered gas cost, with the shipping cost amounting to a little less than 40%. As total annual delivery is doubled, the liquefaction component is decreased by about 15%.

Therefore, the "annual capacity factor" has a marked effect on the unit cost of liquefaction and shipping. As stated before, the effect of increasing shipping distance results in a linear increase in the total cost of gas and the shipping cost component.

It is possible that these costs will decline as a result of the continuing developments in liquefaction plants, storage containers, cargo tanks, and further research. As larger tankers are built, the cost decline will follow the pattern of the earlier supertankers used by the oil industry. It should be cautioned that each project has to be evaluated individually and that it is not possible to generalize the delivered costs for given capacities and distances. As a rule, if the delivered cost of LNG exceeds 60¢/million Btu, it is very unlikely that LNG will be able to compete effectively with existing fuels in the importing country. It is evident, therefore, that the base cost of gas at the source must be sufficiently low to offer a competitive delivered price. For instance, gas costing 10¢/1000 CF in Venezuela could be delivered to the East Coast of Canada in quantities of 350 million CF/day for about 46¢/1000 CF, whereas for the same quantity of delivery from Libya the base gas cost would have to be about 6¢/1000 CF to equal the delivered price. For import of gas from Venezuela to Philadelphia, the delivered price would be approximately 42¢/1000 CF for the same daily quantity.

Markets for LNG

The development of LNG has so far been directed only toward serving a basic fuel market. For baseload LNG supply, the gas has been used either to supplement or to augment dwindling local supply. In some instances the expanded demands for energy in the economy of a country have been met by a new fuel, LNG, instead of conventional fossil fuels. In the United States, where natural gas has established itself in the patterns of energy consumption, LNG has served essentially for peakshaving purposes. Apart from these two main usages, only a limited effort has been conducted on applications that benefit from LNG's unique properties. The markets for LNG can, therefore, be broadly categorized as follows: base load, peakshaving, and other.

The "other" uses-category includes possibilities such as mobile fuel applications in rail, road, air, and water transport, and chemical and petrochemical production. The markets for LNG as a fuel in the base-load and peakshaving applications can be estimated based on projections of energy consumption for individual countries, but a market analysis of the "other" applications for LNG can only be speculative in nature. This is mainly because the technology of these applications is not sufficiently developed and public acceptance will require time before the market can be established.

Estimates of import gas requirements by selected countries in the next decade have already been presented. LNG imports will occupy a fair share of these total requirements. Thus, about 50% of France's imports will be met by LNG (200 billion CF). The estimates on LNG imports in the next decade for some of the countries presented in Table 3 are as follows:

Country	LNG Import, 10^9 CF/yr
France	200
Italy	180
Spain	175
Yugoslavia	30
Japan	300
U.S.A.	180

According to one estimate, by the turn of the century approximately 1 trillion CF/yr of gas will be transported internationally as LNG.

Since the U.S. is the only country with a highly developed gas economy, its peakshaving problem is more acute. Thus, LNG has found a ready application in the U.S. Over 20% of total gas sales in the U.S. can be attributed to peakshaving purposes. Presently, total peaking capacity either installed or under construction amounts to about 15 billion CF. In the next 10 years, the demand for peakshaving gas in the U.S. will amount to over 4 trillion CF/yr. A study of the peakshaving requirements in the U.S. conducted at IGT estimates that about 200 billion CF of this peakshaving demand will be met by LNG, or more than 10 times the currently installed and planned LNG capacity.

The types of large-scale LNG usage for fuel can be classified as follows: domestic and commercial heating, power generation, and industrial process heating. Some of the markets for the last two categories can be sufficient to justify an LNG import project. For instance, electric power generation can be a significant and growing market for LNG. Tokyo Electric Co. has already contracted to purchase liquefied natural gas for power generation to alleviate a serious pollution problem. The Department of Health, Education and Welfare of the United States is currently conducting studies to determine the feasibility of importing LNG for the same purpose to the East and West Coasts of the United States. Many factors, in addition to just the CIF cost of LNG, can influence the selection of LNG over other alternative fuels: The cost of sulfur removal or sulfur control using other fuels, the possibility of lower cost plant construction using LNG, and cost reductions that may be achieved through the use of LNG refrigeration can be important factors that increase the attractiveness of LNG as a power generation fuel.

A whole new major market area, the mobile fuels market, could develop and provide a large and quite lucrative outlet for LNG. At present in the United States and other highly industrialized areas, mobile vehicles are one of the major sources of air pollutants. LNG offers the potential to greatly reduce these pollutants at costs fully competitive with present fuels. Some test work has already been done in the United States on automobiles using liquefied natural gas; the indicated reduction of pollutants is quite significant. High-performance aircraft can derive specific technical benefits by using the refrigeration in liquefied natural gas to absorb the frictional heat generated by their high operating speed. Although the use of LNG in aircraft appears remote today, its usage in cars and fleet vehicles has already been successfully demonstrated. Some idea of the potential market may be obtained from the statistics cited for Los Angeles. It is estimated that, if the City of Los Angeles converted its fleet of 3000 government-owned vehicles to LNG, about 300 million CF of LNG would be required yearly.

Petrochemical and other chemical production can also be an important outlet for LNG supplies. The basic economic considerations of the relative cost of transportation of LNG or the finished products is quite often the determinant in the selection between transportation of LNG from the source to the point of manufacture rather than conversion of the gas to chemicals at the gas fields.

The cold available from the regasification of LNG can be utilized in the liquefaction of industrial gases and to replace refrigeration steps in refinery and petrochemical processes. From a technical and economic viewpoint the best prospects for LNG cold utilization appear to be the following:

1. Liquid air separation
2. Helium extraction
3. Low-temperature distillation of ethane and propane from natural gas
4. Ethylene and propylene stripping from refinery feedstocks

The market for industrial gases shows a very healthy growth rate of 10%/yr. It is estimated that by 1975 the cryogenic gas production in the U.S. alone will exceed 800 billion CF. This is broken down as follows:

Industrial Gas	1975 Production, 10^9 CF
Oxygen	475
Nitrogen	335
Argon	4
Helium	2
	816

Most of the markets discussed above are not only possible but the technology has been developed so that the LNG salesman can get a part of them. However, there are still further possibilities if one considers such applications as fuel for aircraft such as helicopters and subsonic jet airplanes.

The light weight of LNG makes it an attractive fuel for consideration in vertical take-off helicopters (VTOL) and short take-off (STOL) aircraft, ground-effect machines, and hydrofoil boats. The U.S. Navy is presently inviting proposals for a study of LNG use in a "flying-crane" type of helicopter. This is a relatively simple application for LNG compared to a supersonic transport. Aerodynamic drag is not important, and fuel could be stored in lightweight cylindrical tanks. Since no high-altitude operation is involved, fuel boil-off during flight is less of a problem. Although the primary interest in this application may be military, some experiments using helicopters to load and unload ships suggests that this could become a commercially attractive operation, particularly in harbors where ships may be forced to wait for dock space. However, this type of usage is not sufficiently developed to warrant estimation of potential markets.

Conclusions

We have examined the role that LNG can play in meeting the energy requirements of the world. While it has been recognized that LNG has a place in the worldwide energy economy — supplying large quantities of energy to otherwise deficient countries — only a limited effort to find applications that benefit from LNG's unique properties has thus far been undertaken. Many of the specialized markets that take advantage of these unique characteristics of LNG are only now beginning to emerge. Meanwhile, in the immediate future LNG will continue to meet the fuel demands of energy-hungry countries. By the end of the century, LNG will possibly meet 1-2% of the total world energy market. With continued research and development effort in the LNG field, the market for LNG will increase as new uses for this versatile fuel are found and proved.

THE PROSPECTS FOR LNG PEAK SHAVING IN EUROPE

By B.A.F.McLellan and A.E.Twycross

Introduction

The gas industry is highly capital intensive in all phases of its operation from production through transmission to distribution. It is thus elementary economics that to obtain as low unit costs as possible the maximum use must be made of the capital facilities. Peak shaving by liquefying and storing natural gas in periods of low demand, for vaporization and use at the peaks, is probably the most recent innovation to achieve this end.

We shall not be discussing base load supplies of LNG, but only those plants constructed to liquefy, store, and vaporize natural gas specifically for peak shaving.

To this date, some 8 LNG peak shaving plants are being operated in North America by both distribution and transmission companies, and a further 8 are under construction. This number, however, must be related to the volume of natural gas being sold and to other methods of peak shaving in use.

Table 1 gives some statistics comparing the USA market with the European market for natural gas as they were at the end of 1967.

Table 1

		U.S.A.	Europe
Total sales of natural gas 1967	10^9 Nm^3	446.9	25.4
Volume of underground storage			
Total capacity	10^9 Nm^3	128.8	7*
Working volume	10^9 Nm^3	60	2.7*
Volume of LPG/air peak shaving storage in natural gas equivalent	10^6 Nm^3	0.462	N.A.

* estimates.

(Refs., 1,2,3)

However, to consider the future for LNG peak shaving in Europe, we must also look at other factors, and later in the paper the situation in Europe and North American will be compared in the following spheres:

1. The economic environment
2. Geographical enviroment
3. Climate
4. Cost of gas
5. Penetration of gas for home heating and sales volumes.

LNG Peak Shaving for the Distributor

By distributor, we mean the distribution entity which purchases natural gas (usually on a two-part demand/commodity tariff) and distributes and resells the gas to industry, commerce and the residential sector. This is usually the last undertaking in the chain from well-head to ultimate consumer.

The distributor will only undertake peak shaving if by so doing he lowers his overall costs. If he purchases gas on a demand/commodity tariff, he pays not only for the gas itself (the commodity charge) but also the maximum demand requirement he puts upon his supplier. If his demand or peak requirement is less because he has shaved his peaks, his cost of purchased gas will be less. In whatever manner he shaves his peaks it will cost him money, and this cost must be set against the savings in the costs of gas supply; clearly only if his savings are greater than his costs, will he consider peak shaving.

Any assessment or analysis of the best and most economical method of supplying future gas demands will only be as good as the estimate of the magnitude of that future demand. This load forecasting must be as accurate as possible and must take into account the influence of temperature on the temperature sensitive part of the load, the future sales of appliances, their type and unit consumption, the sale of firm gas to industry etc.

Familiarity with the degree day concept is assumed, and the fact that, for long term forecasting, a straight-line relationship between temperature sensitive gas demand and degree day deficiency is accurate. Figure 1 is based on temperature data for Holland (De Bilt Observatory), and also gives a temperature frequency curve for the average year in Essen based on 30 year data.(4) This also shows how the average year on which we must base our economics, bears little relation to the coldest or design year. The term design year is used because if the distributor is to construct LNG peak shaving plant, he must install storage of sufficient size to meet demand in the coldest year, or be left with insufficient gas supply.

In the Appendix an expression is derived to demonstrate the parameters which are relevant for a distributor when considering LNG peak shaving. This expression is as follows: (See also Figure 2)

If $A.S.K. - D.S\,(C+U.R.(M-.5)) > Y.I.$, then peak shaving is economically attractive.

Where A = demand charge in $/yr/Nm^3/day of demand.

S = volume of temperature sensitive gas Nm^3/degree day.

K = $(d_{un} - d_{sh})$ = temperature span over which peak shaving is employed.

d_{un} = degree dificiency for unshaved purchase.

d_{sh} = degree deficiency for shaved purchase.

D = day degrees in average year over the peak shaving temperature range.

C = variable cost of liquefaction and vaporization (in/out cost) $/Nm^3

U = commodity cost of purchased gas $/Nm^3.

R = interest rate applicable to gas stored and not used in average year %

M = a multiplier for D which gives day degrees in coldest year over the same range as D.

I = total investment in LNG plant $

Y = yearly fixed charge on capital employed, plus fixed operating and maintenance costs %

By giving values to the above expression, it is possible to arrive at conclusions as to which parameters have the greatest influence and should be more closely studied. Even without so doing the following conlusions can be drawn:

1. The size and load factor of the base load have no bearing on the savings on purchased gas.

2. The magnitude of the saving in purchased gas is in direct ratio to the magnitude of the demand charge.

3. The cost of peak shaving is in almost direct proportion to the plant investment. Reduction of investment is thus of prime importance.

4. All other things being equal, the total of savings less variable costs is directly proportional to the sendout per degree day (S) for any temperature span (K).

5. The magnitude of the savings in purchased cost of gas is independent of any volume of gas except the sendout per day degree (S). Thus, the saving in purchased cost of gas in any given year, whether average or coldest, is the same, since K is the same.

To look more closely at these relationships, a number of cases have been worked using the following typical costs:

A = $1.0/year/Nm^3/day of demand

U = $0.014/Nm^3 commodity cost

C = $.0042/Nm^3 as in/out cost

Y = 15% which includes 3% for fixed operating and maintenance costs

R = 0.07 as interest rate applicable to gas in storage

The Essen temperature curve of Fig 1 has been used for assumptions of the values of D and K. The capital costs of LNG plants have been estimated from figure 3.

The results are shown graphically in Figure 4, from which further conclusions may be drawn. For example, it is apparent that one very significant factor is the relationship between the design volume and the average volume (or N). Another point that becomes apparent is that the magnitude of the send-out per degree day must be relatively large for LNG peak shaving to be economic. Finally, the ratio D/K must be fairly small, probably below 10 days, before LNG peak shaving could be considered for the European distributor.

A comparison of overall costs against savings is not the only criterion to be considered. The distributor should ensure that storage gas has a lower cost than pipeline gas for that increment of demand at the temperature at which peak shaving is started. In other words, the boundry cost for storage gas should be less than pipeline gas at the LNG/pipeline supply boundary.

This can be ascertained by establishing a tariff for the peak shaving plant and comparing it with the pipeline tariff at that increment of demand. Ideally the peak shaving tariff should be a three part tariff having a Volume Demand Charge, Daily Demand Charge, and In/Out Charge.

This can be reduced to a two part tariff for a specific case. For example, the tariff for an LNG plant costing \$3.7 million (18.7×10^6 Nm^3 storage volume) with 720,000 Nm^3/day peak shaving demand can be approximated as:

$$\frac{\text{Fixed yearly costs}}{\text{Design sendout capacity}} = \frac{3.7 \times 10^6 \times 0.15}{720{,}000} = \$0.77/yr/Nm^3/day$$

plus a commodity cost = (pipeline commodity + in/out cost) \$ /Nm^3.

The storage and pipeline tariffs are equated to establish at how many days use the cost of gas from both supplies is equal:

$$\frac{1.0}{N} + 0.014 = \frac{0.77}{N} + (0.014 + .0042) \quad \text{using previously assumed costs}$$

$$N = \frac{1.0 - 0.77}{.0042} = 55 \text{ days}$$

Thus if this plant were designed to begin peak shaving at a temperature occuring on less than 55 days during the statistically average year (e.g. + 2.3°C using the Essen data from Fig.1.), then the boundary cost of storage gas would be less than pipeline gas.

It is clear that the days use at which the boundary costs are equal is in direct relationship to the magnitude of the variable liquefaction/vaporization cost, but we would point out that this cost is relatively unimportant in the comparative economics of the expression derived in the Appendix unless the ratio D/K is greater than 10-12 days.

LNG Peak Shaving for the Transporter

Peak shaving for a transmission company has different criteria from a distribution company. The principal aim is still to reduce the cost of gas, this time by lowering the unit cost of transmission. It is apparent that this is achieved only if the peak shaving plant is more economic than either increasing the capacity of the transmission pipeline or other forms of peak shaving such as underground storage.

In Europe, transmission lines are usually constructed to transport the estimated peak load of 5-7 years ahead, allowing for the construction of additional compression during this period. Thus it is only after this 5-7 year period that a method of meeting the growing peak demand becomes a problem which is solved either by the addition of new pipeline capacity, or by the development and construction of storage facilities.

Transmission companies have so far used only two forms of storage; underground storage and liquefied natural gas storage. Underground storage can nearly always store larger volumes at lower cost, and normally LNG peak shaving would only be considered when all underground storage reservoirs within an economic distance had been fully utilised. Figure 5 shows the main transmission pipelines supplying the north east U.S. and the underground storage fields operated by these companies only. This shows 8 fields, but in the states of New York and Pennsylvania alone there are 85 separate storage fields operated by some 18 companies.(6 & 7) This is full use being made of depleted fields. The same is already happening in Germany and Italy and will undoubtedly continue as more fields become depleted.

Where there is no possibility of developing underground storage, LNG peak shaving can certainly be an economic solution. This is confirmed by the British Gas Council's proposal for a plant near Glasgow at the end of the main natural gas transmission system to the North of Britain.

Satellite Storage

This, as the name implies, is LNG storage sited at centres of demand, some distance from the main liquefaction and storage plant. No liquefaction is carried out at the satellite storages, boil-off being sent directly into the distribution system. LNG from the main plant is transported in insulated containers to the satellite storage sites by road, rail, or barge.

The principal advantage seems to be the economy of scale afforded to the liquefaction facilities by having one large unit (the capital cost of liquefaction increases by about the 0.6 power of the capacity increase), making LNG peak shaving viable in areas which could not economically support complete plants. A rough indication of the capital cost saving can be obtained from the relative costs of separate and combined liquefaction facilities.

3 liquefaction plants of 30,000 Nm^3/day liquefaction - $\$3.0 \times 10^6$
1 liquefaction plant of 90,000 Nm^3/day liquefaction - $\$1.6 \times 10^6$

The cost of LNG loading and unloading facilities must be deducted from the savings.

Various costs have been quoted for the movement of LNG by road, but in an integrated system this would probably be of the order of $\$2.80/1{,}000\ Nm^3/100$ Km.(8,9,10) It is worth noting that a capital reduction in the above example, of only $\$1 \times 10^6$ with 10%/yr capital charges gives a saving of $\$8.30/1{,}000\ Nm^3$ over $12 \times 10^6 \times Nm^3$ transported (200 days liquefaction x 60,000 Nm^3/day).

Europe & North America

Let us now compare these two areas:

Economic Environment

In the USA, company economics are governed by either the Federal Power Commission, or State Commissions which fix and allow a utility a return on depreciating investments. Regulation in Europe is usually limited to the nationalised industries such as in Great Britain and France, and to a lesser extent municipal distribution utilities.

A transmission company in the USA., pays the gas producer a flat rate, and has only the indirect incentive of a "take or pay" clause to encourage high load factor operation. The real incentive therefore is market competition, which encourages high load factor transmission and consequent lower unit costs. In Europe we have both market competition and the unregulated profit motive.

The distribution company's incentive for improving its purchased gas load factor is the two part rate which is almost universally applied in both North America and Europe.

We can say then, that the economic environment provides as least as great an incentive for load factor improvement in Europe as in North America.

Geographical Environment

The difference between Europe and North America are quite evident here, notably the very much greater distances for gas transmission in North America, and the necessity for very high load factors to achieve low unit cost of transport. (Cf. Figs 5 & 6). If we look further, however, this is mitigated by two factors. First, by the nature of the terrain, pipeline construction costs in Europe average almost 50% more than in North America, and thus cost equivalent distances are less in Europe than in North America. Second, because of the enormous underground storage developed in the U.S.A., the distance between "well-head" and demand centres is not in effect much greater than in Europe. However, Europe also has underground storage possibilities - often in the form of aquifers - and these are well sited in many instances. With this form of peak shaving, storage volumes are so great that they can be operated most economically by either transmission companies or wholesalers, rather than by distribution utilities.

Climahe

We have noted two factors of great significance which affect the economics of peak shaving for distributors and wholesalers. These are the average year degree days in the peak shaving range compared with the temperature span over which peak shaving occurs, which we have called D/K; and the relationship between the coldest year degree days over the peak shaving range and the statistically average year degree days, or M. Since in the U.S.A., companies ranging from those in the deep south (Memphis, Alabama) to those in the midwest and east coast (Wisconsin, Boston) have constructed LNG peak shaving, it is difficult to establish any pattern. However, we would suggest that in the southern states the value of D/K is fairly low, while in the north although D/K may be higher, M is of the order of only 2. The value of M in northern Europe is nearer 3 or 4 than 2, as may be seen from Fig.1.

Since Europe in general has a higher D/K ratio than the southern states, and a higher value of M than the north east U.S.A., the climatic opportunity for peak shaving by distributors is not as great in Europe as in North America.

Cost of Gas

As we have seen the economics of peak shaving for the distributor are influenced very little by the commodity cost of the gas or the demand/commodity ratio. Economics are however highly dependant on the demand cost in absolute terms.

In the U.S.A., for those distribution companies using LNG for peak shaving, the demand cost of purchased gas supply varies from about \$0.6 to \$2.4 /yr per Nm^3 per day of demand as we move from the southern states to the north east coast.(11) Any comparison is complicated by a "Storage Service" being available to some distributors such as Brooklyn Union and Philadelphia, and a "Winter Service" to Boston. These services effectively reduce the demand charge in the relevant peak periods.

In Europe, wholesalers will pay anything from \$0.33/yr per Nm^3 per day of demand in Belgium, to \$1.15/yr Nm^3 per day of demand near Russelheim in W. Germany. Distributors pay from \$0.65 to \$1.2/yr per Nm^3 per day of demand depending upon their yearly volumes and their distance from the well-head. Many European tariffs are actually based on hourly rather than daily demand, and we have converted these to a daily basis using either 20 or 22 hours depending on the circumstances. Here LNG has the advantage by being able to shave peak hour demands in a way which underground storage further from the demand centre could not.

It would be fair to say that from a tariff viewpoint the European distributor or wholesaler would have as much incentive as his north American counterpart to consider LNG peak shaving.

Penetration of Gas for Home Heating and Size of Utility

We take these two points together because they both give some indication of the send out per degree day, which has a significant bearing on whether LNG peak saving could provide economies for the distributor, and the magnitude of the peak which the transmission company must carry.

The yearly demands of some of the U.S. distributors can be seen from Table 2. It is true that the volume of gas used by industry in some of these areas is large, but nevertheless the domestic and commercial volumes are certainly far in excess of what may be expected by distributors in Europe. Using available data it is possible to estimate the temperature sensitive demand per centigrade degree day as ranging from about 60,000 Nm^3/dd^{o}C for the two smallest distributors to over 300,000 Nm^3/dd^{o}C for some of the larger.

In Europe one of the larger distributors is Hamburger Gas Werke which has a yearly demand approaching 350 x 10^6 Nm^3 in natural gas equivalent, and a temparature sensitive demand of over 50,000 Nm^3/dd^{o}C in natural gas equivalent for about 450,000 connections.(12) The Scottish Gas Board has a yearly demand of nearly 800 x 10^6 Nm^3 natural gas equivalent for about 970,000 connections.(13)

These European figures are for 1967/68. In the future, as natural gas penetrates further and large areas become converted to 100% natural gas with consequent reduction in end user tariffs, we can expect these figures to increase rapidly.

Conclusions

A combination of low demand per degree day and large design/average storage capacity ratios make it appear unlikely that European municipal distributors will find LNG peak shaving plants attractive economic propositions within the next few years.

Transmission companies will develop underground storage as a first choice when pipeline capacity becomes fully utilised. There seems scope for this development in Austria, Germany and Italy, considering the number of small field which are being depleted. Other parts of Europe may be able to use aquifers.

The wholesaler who buys from transmission companies and sells to large industry and municipal distributors, has the necessary volume per degree day and tariff incentive for peak shaving, and can reduce his investment in main lines by judicious location of LNG plants. Satellite peak shaving storage supplied from a central LNG plant within the wholesaler's area would appear to be the answer to his problems.

We consider it a valid generalization that satellite LNG peak shaving by the wholesaler is the field of greatest potential growth in Europe, and one which should be more closely studied.

Table 2. LNG peak shaving projects

COMPANY		Storage Volume 10^6 Nm^3	Sendout Capacity 10^6 Nm^3/day	Liquefaction Capacity 10^6 Nm/day	Annual Sales 1967 10^6 Nm^3	Average Degree Days to 65°F	Penetration of gas customers for Heating State Average
Alabama Gas Corp.	*	17.5 +	2.4	0.13	2,230	2,780	93%
Boston Gas Co.	* †	34	3.5	0.17	742	5,634	36%
Brooklyn Un. Gas Co.	†	17.5 +	2.8	0.16	2,410	4,810	35%
Memphin G.L.+W. Div.	*	28	5.6	0.14	2,680	3,116	93%
Northwest Nt. Gas Co.		17.5	1.7	0.07	1,870	4,365	82%
Northern States Power	*	11.3 +	1.0	0.056	1,740	8,000	64%
Philadelphia Gas Works	* †	113	11.3	0.45	2,250	5,144	63%
Tennessee Gas Tr. Co.	†	85	7.1	0.52			
Texas Eastern Tr. Co.	†	57.5	5.6	0.285			
Transco Gas P.L. Corp.	†	28	5.6	0.2			
San Diego G.+El. Co.	*	17.5 +	1.7	0.07	n.a.	1,458	100%
Wisconsin Nat. Gas Co.	*	9	1.4	0.028	1,020	7,200	64%
Cryogenic Ent.							
Vancouver		20	n.a.	0.08		4,900	
Richmond		1.8 +	n.a.	0.01		4,900	
Northern and Central Gas Co. Toronto		17.5 +	1.1	0.07	2,500	7,300	
Quebec Natural Gas Co. Montreal		31	n.a.	0.283			
British Gas Council Lanark		28.3	3.5	0.14	700 (N.G.equiv)	4,435 (to 60°F)	
Stuttgart		20	n.a.	n.a.			

* indicates that distributor also operates LPG/air peak shaving
† indicates that distributor or transporter has available or operates underground storage or peaking service
+ indicates project operational

(Refs.,1,7, 15)

APPENDIX

Assumption: It will not be economic to install LNG peak shaving unless the yearly total costs saved on purchased gas are greater than the yearly total costs of owning and operating the LNG plant.

Yearly total costs saved on purchased gas = (yearly pipeline demand charge for unshaved demand) – (yearly pipeline demand charge for shaved demand)

(Refer to Fig 2)

Let S = volume of temperature sensitive demand per degree day Nm^3/dd

B = volume of base load gas per year Nm^3

d_{un} = degree day deficiency for unshaved purchase demand $^\circ C$

d_{sh} = degree day deficiency for shaved purchase demand $^\circ C$

L = load factor of base load gas in days

Hence $(B/L + S.d_{un})$ = total peak day demand from pipeline for unshaved gas Nm^3/day

and $(B/L + S.d_{sh})$ = total peak day demand from pipeline for shaved gas. Nm^3/day

If A = pipeline demand charge for purchased gas in $\$/yr/Nm^3$ per day demand then the saving per year in pipeline demand charge

$= A\,(B/L + S.d_{un}) - A(B/L + S.d_{sh})$

$= AS(d_{un} - d_{sh})\ \$\ /\ year$

Let $(d_{un} - d_{sh}) = K$

then the saving can be expressed as A.S.K. $\$/year$.

The yearly total costs of owning and operating the plant include

a) the yearly fixed charge on capital employed
b) the variable cost of gas vaporised and liquefied per year
c) the interest charge on gas in storage and unused per year

Let D = degree days in the average year to the peak shaving temperature i.e. the degree days between d_{un} and d_{sh}

MD. = degree days in coldest or design year to the peak shaving temperature where M is some multiplier.

C = variable cost of liquefaction and vaporization $\$/Nm^3$

Π = commodity cost of purchased gas $\$/Nm^3$

R - interest rate chargeable to gas bought but unused in storage %

Y = yearly fixed charge on capital employed plus fixed operating and maintenance costs %

I = capital employed or cost of plant required to liquefy and vaporise gas, and store a volume of M.D.S.

Then total cost per year of owing and operating plant can be expressed by:

Y.I = yearly fixed charge on capital employed plus fixed operating + maintenance costs

plus D.S.C. = variable cost of gas liquefied and vaporised

plus U.R. (M.D.S. - 0.5 D.S.) = interest charge on gas bought but unused.

Note: a half years interest is charged on gas used to allow for the time between buying and using.

$$= Y.I. + D.S. (C + UR (M - ,5))$$

Finally we can say that peak shaving by LNG would be an economic proposition if $A.S.K. - D.S. (C + UR (M - .5)) > Y.I$

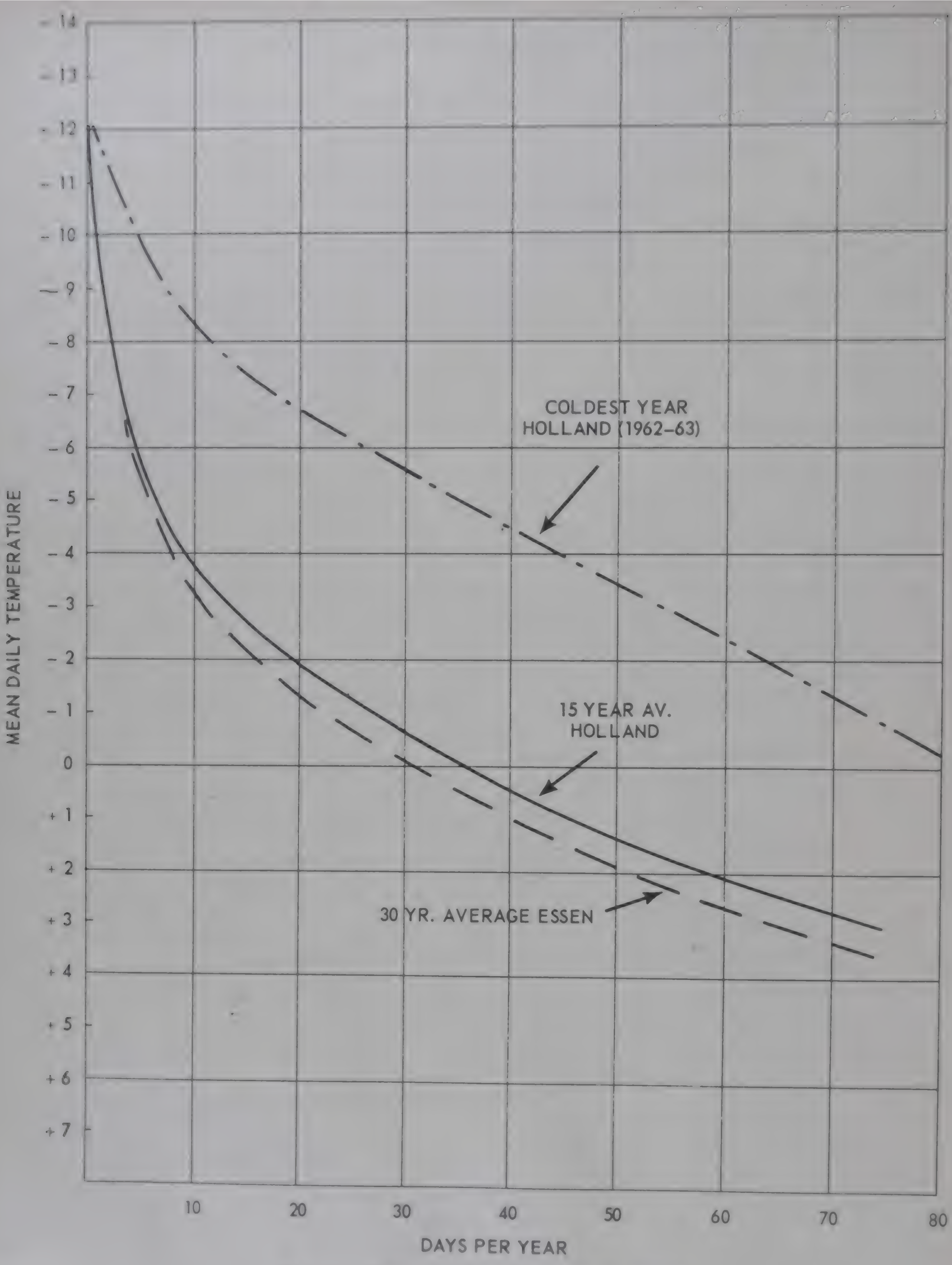


Fig.1. Temperature frequency curve

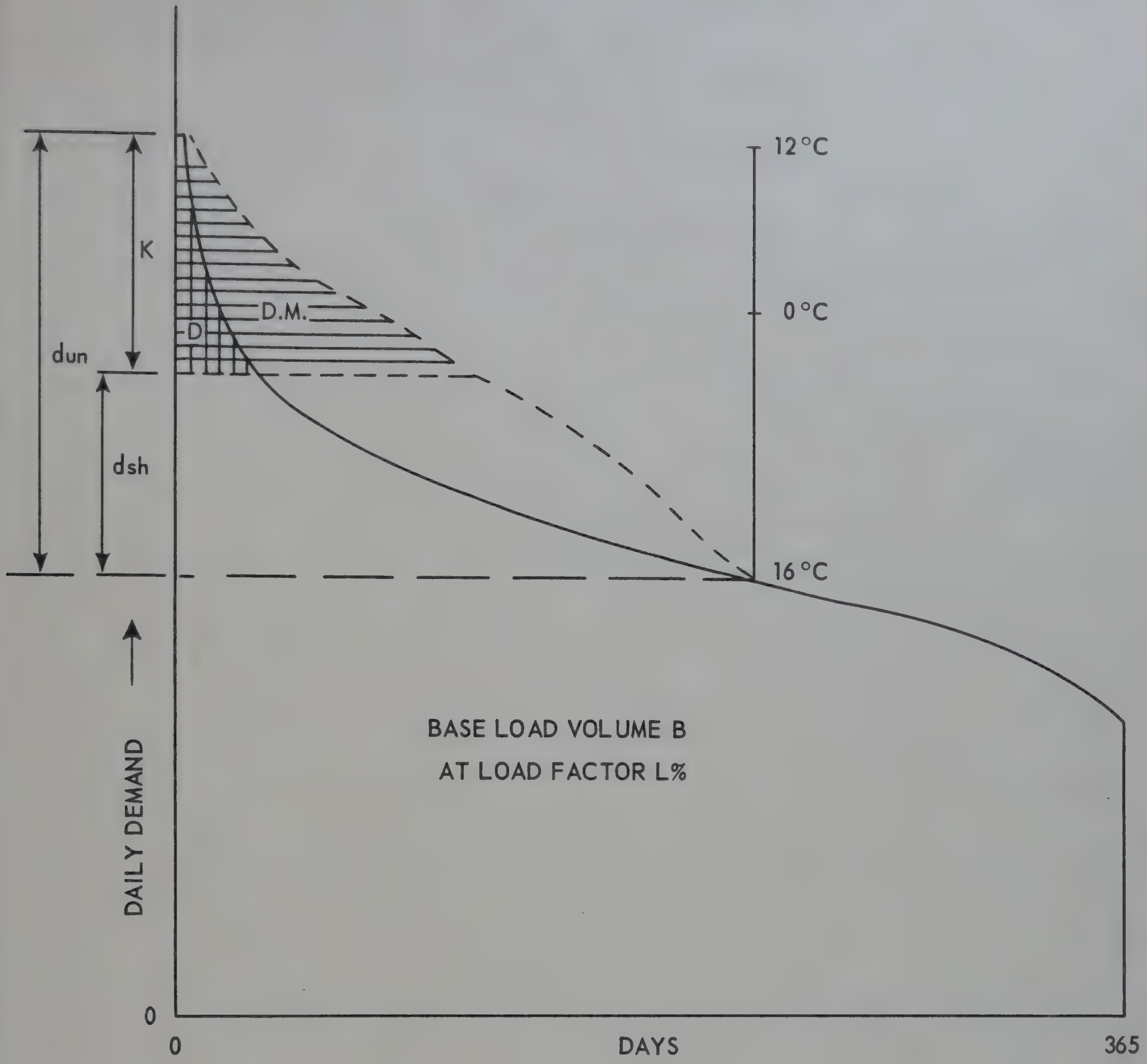
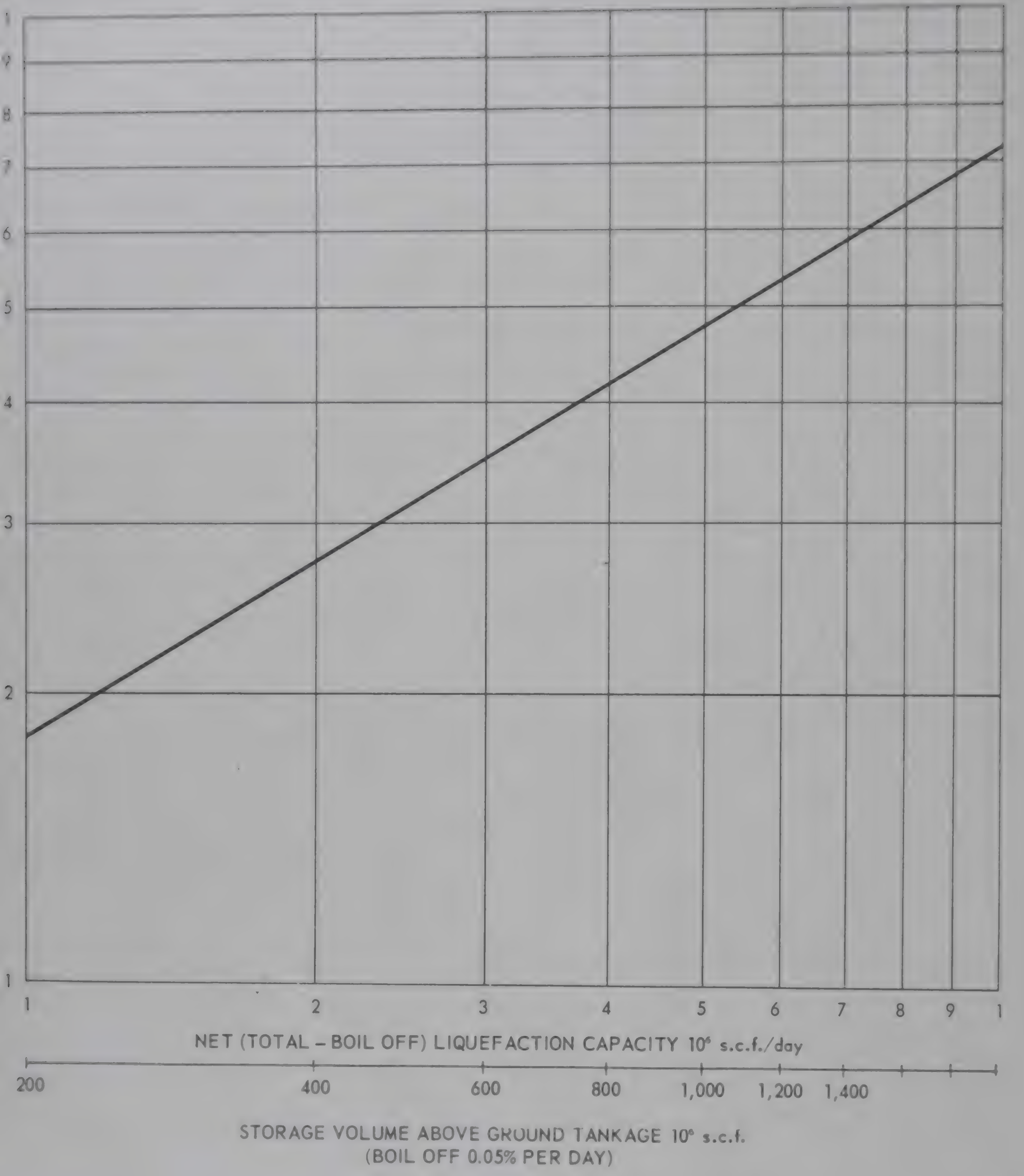
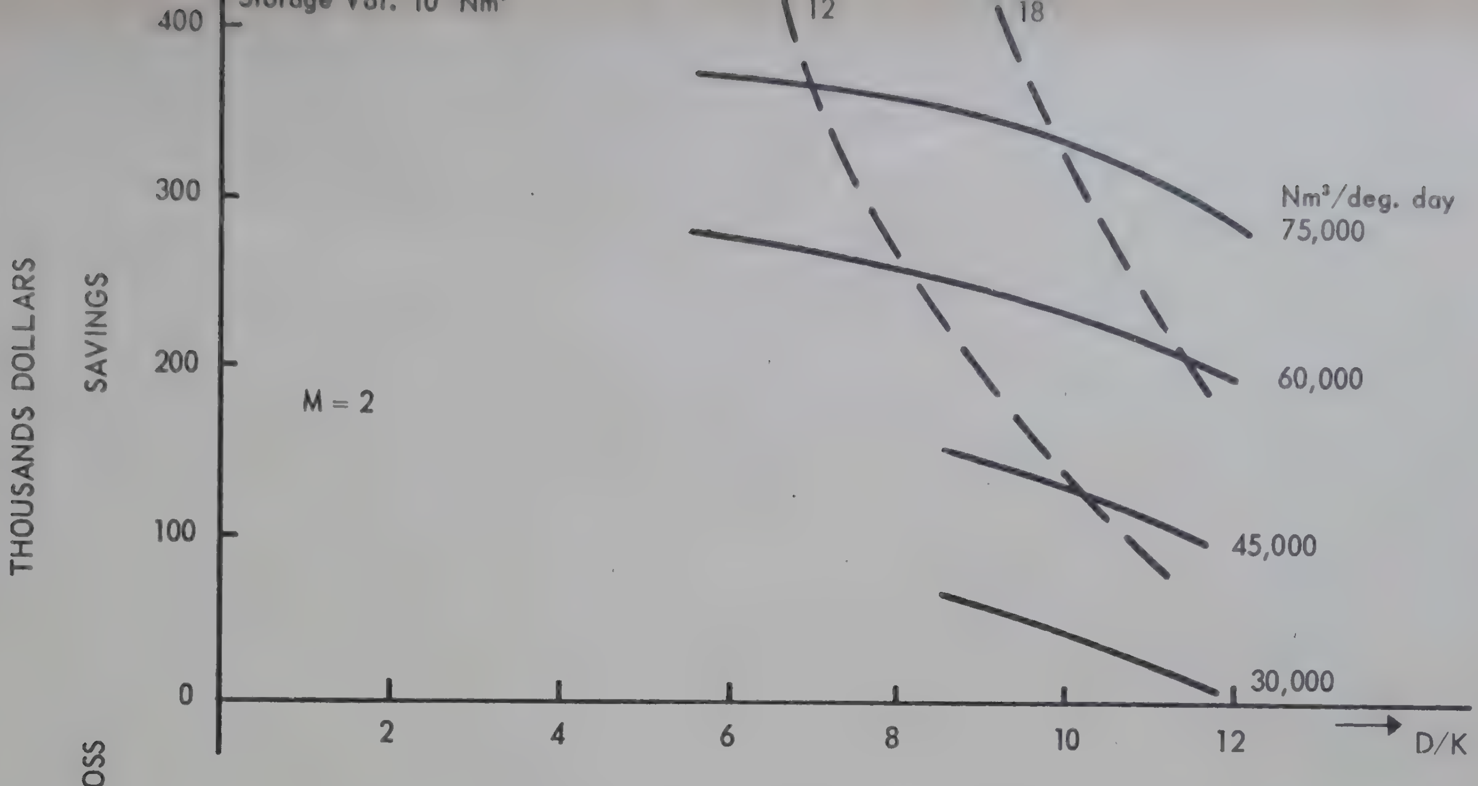

K
dun
dsh
D.M.
D
12°C
0°C
16°C
DAILY DEMAND
BASE LOAD VOLUME B
AT LOAD FACTOR L%
0
0
DAYS
365

Fig.2

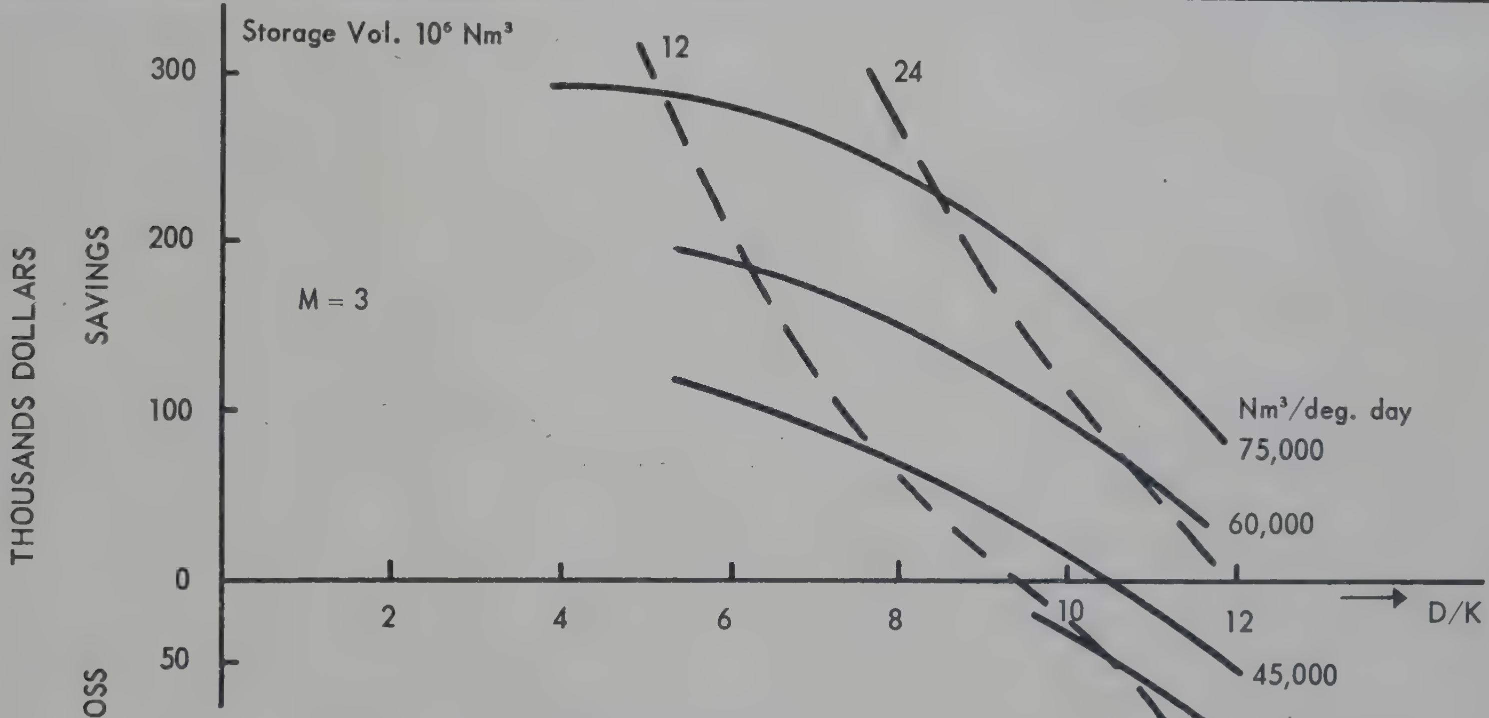


(after Kennedy, Oil & Gas Journal 18 Sep. 1967)

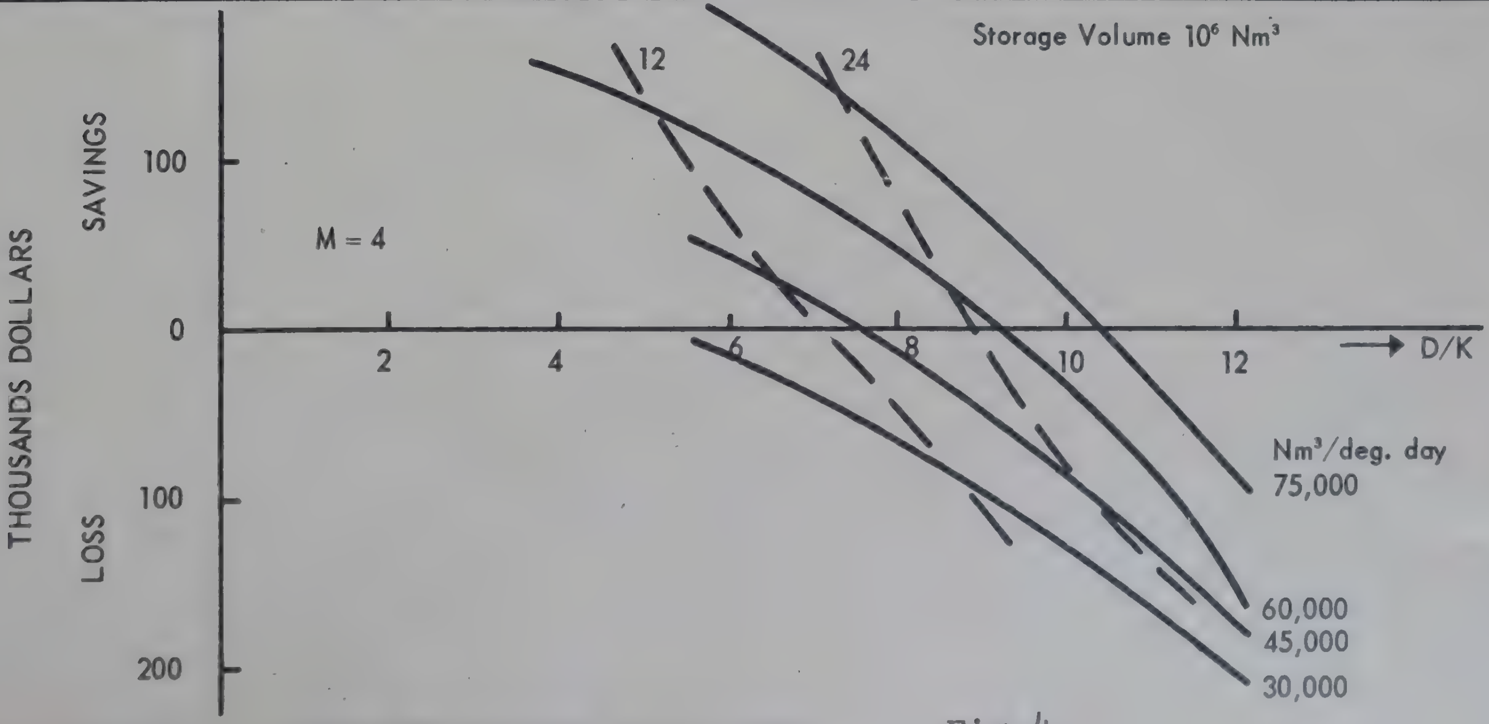
Fig.3



COLDEST YEAR VOL. 2 x AV. YEAR



COLDEST YEAR VOL. 3 x AV. YEAR



COLDEST YEAR VOL. 4 x AV. YEAR

Fig.4

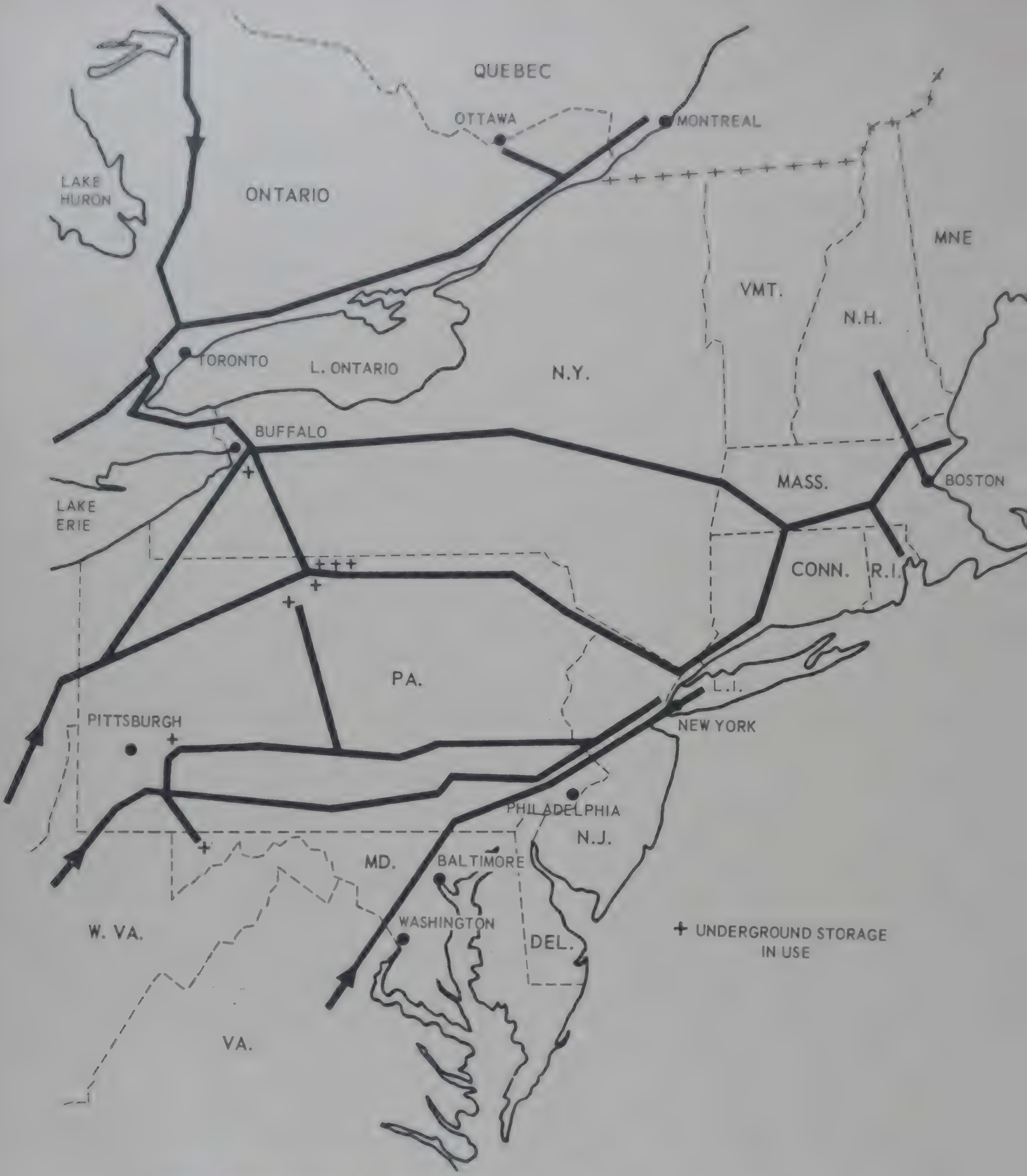

QUEBEC
OTTAWA
MONTREAL
LAKE HURON
ONTARIO
MNE
VMT.
N.H.
TORONTO
L. ONTARIO
N.Y.
BUFFALO
MASS.
BOSTON
LAKE ERIE
CONN.
R.I.
PA.
L.I.
PITTSBURGH
NEW YORK
PHILADELPHIA
N.J.
MD.
BALTIMORE
WASHINGTON
W. VA.
DEL.
+ UNDERGROUND STORAGE IN USE
VA.

Fig.5. Main natural gas transmission lines - N.E. U.S.A.

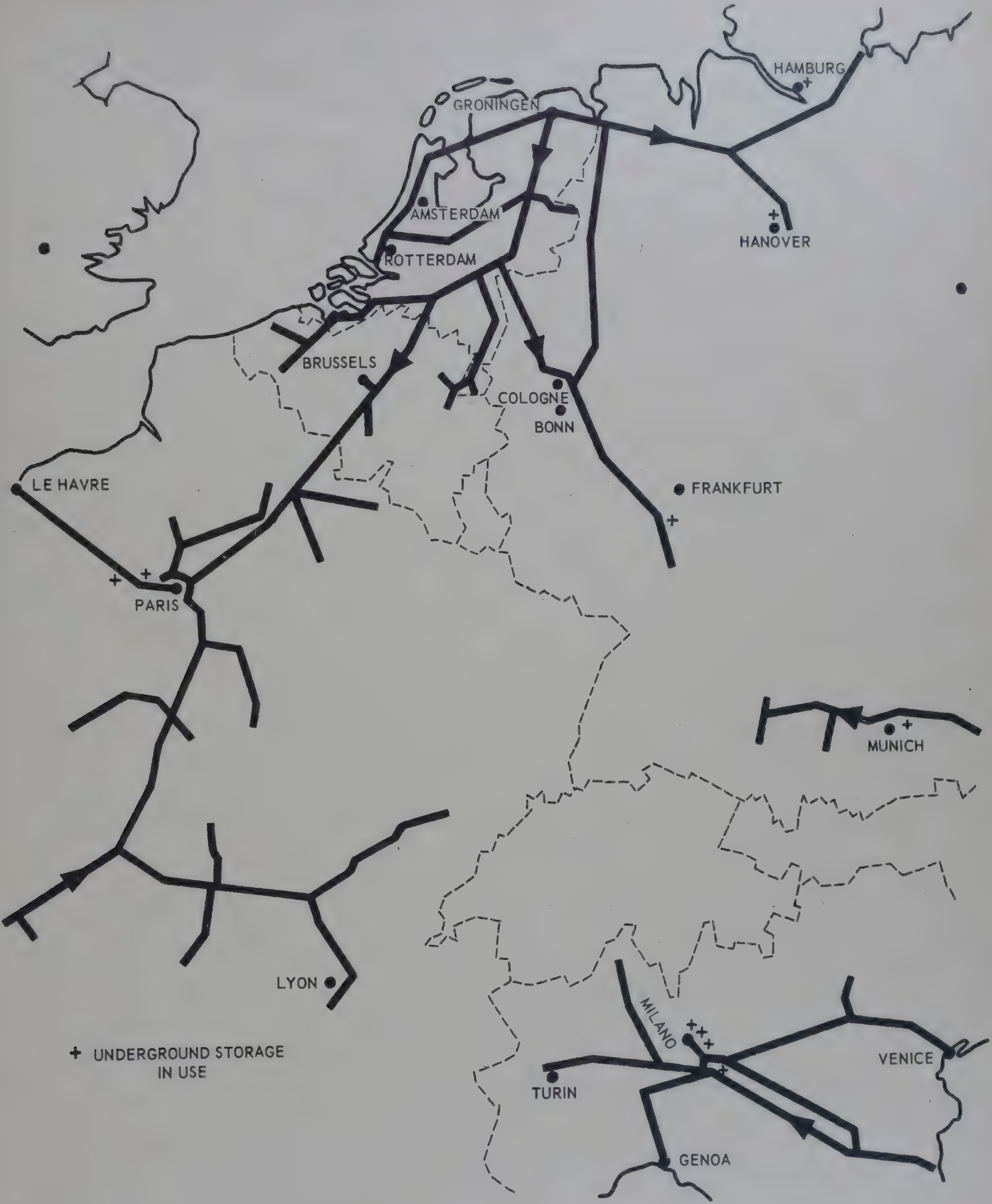

HAMBURG
GRONINGEN
AMSTERDAM
HANOVER
ROTTERDAM
BRUSSELS
COLOGNE
BONN
LE HAVRE
FRANKFURT
PARIS
MUNICH
LYON
MILANO
VENICE
TURIN
GENOA
+ UNDERGROUND STORAGE
IN USE

Fig.6. Main natural gas transmission lines - N.W.Europe

HOW IMPORTED LNG WILL COMPLEMENT DOMESTIC GAS

By B.de Frondeville, J.T.Jensen and K.H.Wulff

Introduction

Liquefied natural gas is becoming increasingly important as a storable form of natural gas (1/600 reduction in volume at -260°F and atmospheric pressure) either for shipments from energy-surplus areas of the world to energy-deficient areas, or to meet peak demands by distributing utilities. Two prototype LNG ships ply the ocean in the LPG and ethylene trades respectively, three larger carriers are operating regularly out of Algeria (1, 2), six more are being built to haul LNG from Libya (3) and Alaska (4), and five others are in the bidding stage. Thirteen peakshaving plants in the U.S. and six elsewhere in the Western world have been built or contracted for since 1965.

Ship design is receiving a great deal of attention from both economic and safety standpoints (5,6), and ship capacities are being substantially increased to reduce costs. Unfortunately, the large volume of LNG that must be shipped for economical international trade (at least 100 MMcf/d) limits the immediate market: few areas can accept such large shipments without disrupting their energy patterns. Other limitations on the potential for LNG as an international commodity are the success of offshore explorations, declining costs of nuclear power and oil desulfurization, and the development of nuclear blasting and other methods for obtaining new gas resources. Thus, the present pattern of individual shipping arrangements, taking advantage of special situations, should continue.

However, discovery of offshore and other natural gas resources in energy-deficient countries will open or extend the market for gas and improve its future competitive position in relation to other energy sources. Then, as nearby reserves fail to keep pace with growing markets, LNG will be a logical choice to enter these markets on a broad basis.

The situation may soon develop in the United States. The declining domestic reserves-to-production ratio has led several utilities to study natural gas imports as future insurance. LNG has been considered because of various limitations on Canadian and Mexican imports by pipeline. Of the two major coastal areas of interest, USNH (North of Hatteras) and the West Coast, the latter offers little promise for importation during the 1970's because:

- West Coast peak loads are shorter and more erratic.
- Underground storage is available near centers of load.
- Present supply lines will be extended.
- Additional Canadian gas will be permitted to enter.
- Reserves of low-cost West Texas gas are adequate.
- New reserves have been found off the California shore.
- Accessibility to low-cost supplies of LNG is poor.

On the other hand, two spot LNG shipments have been made to a Boston Gas peakshaving plant from Algeria, in the 1968-69 winter season.

This paper therefore concentrates on LNG imports to the U.S. East Coast and examines why and how LNG could be successfully imported under present conditions.

WHY IMPORT LNG?

Domestic Supply/Demand Balance

A recent energy forecast by the Office of Oil and Gas, Department of the Interior concluded that sufficient supplies of domestic natural gas will be developed to meet future requirements and sustain a 15:1 ratio of reserve to production in 1980. These conclusions would no longer hold if a more optimistic gas demand forecast were used, such as those recently published by Texas Eastern (8) and the Future Requirements Committee (9).

Figure 1 shows the effect of a higher rate of demand growth on the reserve-to-production ratio. The "alternate forecast" assumes that total U.S. natural gas demand will grow slightly less than 5% per year, compared with historical growth rates of slightly more than 5% per year for the past ten years. This forecast is consistent with that of the Future Requirements Committee through 1970 but is more optimistic thereafter. On the supply side, the alternate is based on a Department of Interior projection of reserve additions (10) similar to that used by the Office of Oil and Gas.

A reserve-to-production ratio of 10 in 1976 will not necessarily produce significant shortages if national demand can draw on all supplies. However, the increasing tendency for Texas supplies to be reserved for the Texas local market will accentuate a tight interstate supply/demand balance and lead to an interstate reserve-to-production ratio closer to 8 than 10.

A domestic gas shortage before 1980 is a serious enough possibility to stimulate research and development of several unconventional means of increasing supply--nuclear fracturing; coal, lignite, or shale gasification; and importation of LNG. On the basis of actual commercial achievements, LNG has the greatest prospect for immediate significant supply augmentation.

Interfuel Competition and Effect of Air Pollution Restrictions

The energy market on the East Coast can be divided into three categories: the electrical generation market consuming 30% of the gross heat input, the non-utility industrial market consuming 20%, and the residential and commercial market consuming the remaining 50%.

Gas has not achieved a very significant share of the heat input for electrical generation, because utilities are most sensitive to the price of the fuel per unit of heat content. On a pure heating value basis, gas is not competitive with residual oil or coal. Furthermore, electrical generation is considered to be an inferior use of natural gas, and the Federal Power Commission attempts to inhibit sales to this market.

At the other end of the spectrum, gas has taken much of the residential and commercial market where available. The residential consumer of fuel is hardly aware of the relationship of price to heat content; so long as overall economics are not noticeably disparate, he is likely to choose on the basis of convenience and other qualitative considerations.

Among the three types of energy markets, the greatest diversity is found in the non-utility industrial sector. Figure 2 is a simplified representation of the energy conversion economics for such users. The figures on the horizontal scale are a measure of the size of a consumer: the 0.6 point, for example, represents an industrial user whose annual consumption of fuel is such that 60%

of the heat content of all non-utility industrial fuel is consumed by users of smaller size. The intersection of the 0.6 ordinate with the gas curve shows that the cost of gas to such a user is approximately $0.50 per million Btu.

To put the economics of burning coal and fuel oil on a comparable basis, the cost of handling, storage, and energy conversion facilities must be considered. To do this, we have calculated a premium that equates a gas installation to an oil or coal installation on a discounted cash flow, present-value basis. Thus, for a 0.6 user, the fact that the residual oil curve is lower than the gas curve means that the price of residual oil plus a premium that accounts for lesser convenience is still more economical than gas. Similarly, if this user already has a coal-burning installation, he will not find it economical to convert to gas; investment in a new coal installation, however, would be far less attractive. The difference between the new-coal curve and the conversion-coal curve reflects the costliness of new coal burning installations for this size of user; if his facilities are already built, on the other hand, his investment in them is a sunk cost and does not enter in the premium.

Figure 2 shows that gas is indeed most economical for those smaller users responsible for some 30 to 40% of industrial fuel consumption. Larger users find residual oil more attractive. Nevertheless, coal is still economical in many cases and provides about one-third of the heat used by non-utility industries. This is because the curves represent average values; they would be more realistically shown as bands that overlap when they approach each other. During the past five years coal consumption has been declining in this industrial market while oil has been growing slightly and gas has been growing faster.

Because importers may bring in residual fuel oil in unlimited quantities to the U.S. East Coast, its price in the East approximates that on the world market rather than that associated with costlier U.S. crude. Much of this oil comes from Venezuela and is high in sulphur content. The removal of this sulphur would cost customers an estimated additional 30 to 40¢ per barrel, or 5 to 6¢ per million Btu. In fact, the oil companies have already increased their prices of low-sulphur fuel oil to 37 or 38¢ per million Btu--slightly less than the price of natural gas.

Air pollution restrictions will thus have the effect of shifting the coal and oil curves upward for the increased cost of either low-sulphur fuels or stack gas removal of sulphur dioxide. Since 1966, the period on which Figure 2 is based, the upward shift is represented by the dotted lines that parallel these curves. Residual oil has remained slightly more attractive than gas, but its competitive margin has narrowed. While existing coal users have the same incentive to convert to oil, the attractiveness of gas has substantially improved.

As the burning of coal in existing facilities becomes more uneconomic, many will switch to gas; the competitive edge of residual oil will be quite small unless one of the Maine refinery projects (Occidental, Shaheen) are implemented. Nevertheless, air pollution restrictions will not open up massive new markets for gas because of the availability of low-cost imported or free-trade-zone fuel oil.

HOW LNG WILL BE IMPORTED

Sources

The geographical location of known reserves of natural gas, whether associated or not, suggests that LNG imports may come from the following countries, in order of increasing distance:

Venezuela and the Caribbean, 1800 to 2000 miles from USNH*

North Africa, essentially Algeria (3950) and Libya (4700 miles)

Nigeria, 5000 and 5200 miles from Boston and Philadelphia respectively.

Production and reserves in these countries are listed in Table 1.

Nigeria had been sedulously considered as a source of LNG for the United Kingdom prior to the discovery of North Sea gas, but it is 300 to 1100 miles farther than North Africa, with probably higher construction costs onshore, and will take some time to recover political equilibrium.

Although Libya's potential reserves are not yet fully assessed, they are clearly much smaller than those of Algeria and farther away; thus, Southern Europe would be a more natural market than the U.S. East Coast. Political considerations might influence U.S. importers, however, so we have not excluded Libya from our economic analysis.

Algeria has a 700-800 mile advantage over Libya. Its Hassi R'Mel field is among the largest in the Western world but is some 400 miles from the coast.

Venezuela has the largest reserves in the Caribbean. Colombia, Mexico (which is already exporting some gas to the U.S. by pipeline), and Trinidad have much smaller production potential.

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*NH = North of Hatteras

TABLE 1

CHARACTERISTICS OF POTENTIAL LNG SOURCES TO USNH

	Reserves (Tcf)	20-Yr. Potential (MMcf/d)	Production 1966 (MMcf/d)	1972 Committed Gas Local & LNG (MMcf/d)	1972 Available for Exports (MMcf/d)
Venezuela	32	3,200	4,100[a]	4,000[b]	1,000[e]
Algeria	80	8,000	830[d]	750	7,250
Libya	15	1,500	800[c]	500	1,000
Nigeria	4	400	250[c]	50	350

[a] 37.1% flared, 45% reinjected, 9.6% industry auto consumption, 7% sold, 1.5% transformed.

[b] 80% of expected production, including 55% for reinjection.

[c] Almost totally flared.

[d] About 68% flared, 20% exported, 12% consumed in Algeria.

[e] Gas being flared according to plan; second column shows the need for new discoveries or development to sustain long-term contracts.

As shown in Table 1, however, the long-term availability of Venezuelan gas for exports is questionable unless large reserves of non-associated gas are discovered or developed. A fast-developing reinjection program and a strong emphasis on local development of industry should steadily reduce the amount of flared gas, currently some 1500 MMcf/d. LNG exports appear to have little priority, particularly because they might cut into Venezuela's sizable oil and fuel exports to the United States.

Importation Costs

An LNG importation project includes a number of steps: production and gathering of natural gas, liquefaction and storage at a seaport, shipment by specialized tankers, storage at the receiving terminal, and regasification for distribution to consumers.

Our analysis is based on a natural gas cost of 10¢/Mcf at the inlet of the liquefaction plant; transportation costs from remote inland locations might raise this value unless offset by by-product credits. Liquefaction and loading terminal investment including port facilities have been costed as grass roots installations, although small credits were taken in North Africa for existing export terminals.

For these particular calculations, ships fitted with membrane "integrated" tanks (such as selected by Phillips-Marathon and Gazocean for the LNG trade) have been considered rather than the more conventional self-supporting tanks or a partly pressurized system. Finally, the East Coast terminal has been priced in the Philadelphia-New York area.

A capital recovery ratio of 15% was used throughout the LNG shipping project. This is a minimum value, considering the risk of the venture: a substantial part of the investment layout is tied to the foreign source, and the ships are so specialized as to be practically bound to a particular project--at least for the next decade. Table 2 and Figure 3, which summarize our estimates and calculations, are in general accordance with the numbers proposed by Gertz and Wetzel (11). They yield a cost of about 5¢/Mcf for each additional 1000 nautical miles of distance.

TABLE 2

COST ESTIMATES FOR IMPORTATION ON U.S. EAST COAST

	Venezuela (1900 n.mi.)			Algeria (3950 n.mi.)		Libya (4700 n.mi.)	
Volume Landed (MMcf/d)	150	300	600	150	300	150	300
Number x size ships (m^3)	95,000	2 x 95,000	4 x 95,000	2 x 87,000	3 x 115,000	2 x 102,000	4 x 102,000
Cumulative Investment (MM$)							
Liquefaction, Marine Facilities, Storage	64	94	153	68	101	70	105
Ships	29	56	110	52	96	60	118
East Coast Terminal	14	20.7	31	14	23	15.3	26
Total	107	171	294	134	220	145	249
Landed Costs (¢/Mcf)							
(Inlet plant)*	10	10	10	10	10	10	10
C.i.f., liquid	47.9	41.9	39	60	53	64	57
Gasified	53.2	46.5	43.3	65	57.5	69	61.5
Receiving Storage (Bcf)	6	9	12	6	11	6	12

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*6.5¢/Mcf correction on c.i.f. prices for 5¢/Mcf increment in inlet gas price.

Gas inlet price is a direct part of the sales cost. Taking into account autoconsumption, losses, and inventories, a 5¢ differential in this price would become a 6.5¢ differential for Venezuelan gasified LNG and a 7¢ differential for North African gasified LNG. C.i.f. costs for liquid LNG ex-tanker would support slightly smaller differentials, about 0.2¢ per Mcf lower than the former.

By-product credits have not been taken into account in these calculations. Such credits could stem from the extraction of C2, C3, and C4. Ethane could be recovered for ethylene production at either end, but less cheaply than propane and butanes. However, the exact credit attributable to these products depends much on their local marketability, and the current oversupply of ethylene and LPG in the Caribbean Gulf could last through the early 1970's. The situation is somewhat better on the East Coast, but such a discussion is outside the scope of this paper. Helium extraction from flash gas (or flue gas) at the liquefaction plant should also be considered if the former is present and marketable. The same considerations apply to recovery of sulfur from sour gas, but contents between 2% and 6% are both costly to eliminate and not very profitable. Finally, nitrogen and carbon dioxide have been considered to be in minimal amounts, as they have to be removed at a cost penalty without much recovery value.

Recovery of refrigeration value at the receiving terminal has been widely investigated. A recent analysis (12) showed that utilizing the -260°F heat sink as a refrigeration source is much more attractive than the extraction of work from a power cycle.

Value for Base Load

The average cost of domestic gas at a 100% load factor in the New York area is 40¢/Mcf, based on the rates shown in Figure 3 and/or transmission companies' statements. Large volumes of LNG imported from the closest source would cost about 45¢/Mcf, and shipping costs would make North African gas 10¢ to 15¢/Mcf higher. The possibility of an interruption in the supply of such large volumes, for technical or political reasons, would also be of great concern to U.S. importers. We must then turn to other means of bridging the price gap between domestic and imported gas.

The by-product credits described above may be substantial, but the many simultaneous conditions necessary to make them fully operative might prevent sound project economics to be solely based on such considerations, at least for those which are not reflected in firm contract inlet prices. They should, rather, come as an added source of revenues, allowing easier access to investment sources for expansion or new projects.

Government regulations--air pollution restrictions, regulation of natural gas sellers as utilities, and the Oil Import Program--increasingly affect interfuel competition in the United States. On balance, they favor increasing penetration of gas into the industrial market or higher prices. Thus, an imported LNG project could take a cost credit that would bring it into closer competition with domestic gas. We estimate that this credit ranges from 3¢ to 8¢/Mcf, which correspond to what stack gas removal of sulfur might cost present coal users. Although this is the range necessary for large LNG imports to be competitive, the higher part of the range would apply most probably to volumes in the order of 150 to 300 MMcf/d.

Such a credit would also sustain the present competitive balance between residual oil and gas for those users who have chosen gas over residual oil in spite of apparent economic penalties.

Peakshaving

To meet widely fluctuating demand and at the same time maximize the utilization of expensive pipelines, transmission companies store gas near the consuming end during off-peak periods. Much is stored underground in Appalachia; the rest is liquefied to take advantage of the reduction in volume and is stored nearer the main consuming centers. We shall consider two ways in which the stored gas is used--seasonal load balancing and needle peakshaving--and assess the economics of substituting imported LNG in each case. (13)

To estimate the cost of gas to pipelines serving the East Coast, we have treated them all as one system whose elements are a pipeline from the Gulf Coast to Appalachia that operates at 100% load factor, underground storage facilities in Appalachia, a 500-mile line from Appalachia to the market center on the East Coast that operates at approximately a 50% load factor and lastly LNG peakshaving facilities on the coast.

Rather than making a complete mathematical model of present East Coast pipeline systems and then expanding it to account for imported LNG projects, we have simplified the calculations by isolating subcomponents of the system based on the load pattern of natural gas consumption on the East Coast in 1966. A rigorous solution to systems optimization would produce a slightly different result.

(1) Seasonal Load Balancing:

Balancing of a seasonal load (1966 East Coast consumption) is illustrated in Figure 4. The triangle represents seasonal load over 270 days of the year; the dashed rectangle indicates the equivalent consumption spread over the entire year at a fixed daily rate equal to maximum demand times the load factor (37%). The economies of transmitting gas at this uniform rate can be achieved if enough gas can be stored during the summer to provide the excess needed in the winter.

Table 3 compares two ways of meeting the storage requirement: 1) piping the gas from the Gulf to Appalachia, storing it underground and piping it the remaining 500 miles to the East Coast during the winter; and 2) importing additional LNG, storing it near the consuming centers, and regasifying it as needed. Costs are calculated for two delivery rates, 150 and 300 MMcf/d.

Considering first the point of view of the pipeline transmission company, we estimate the cost of gas at the 100% load factor delivered in Appalachia at 35¢/Mcf. Costs of underground storage were based on recent storage additions to the Columbia Gas system. Although underground storage saves additional pipeline capacity from the Gulf to Appalachia, the line from Appalachia to the coast still operates at only a part load factor. The economics are believed to have reached their plateau for the volumes considered

TABLE 3

SEASONAL LOAD BALANCING

(V = 150 or 300 MMcf/d)

Gulf Supply		
Gulf to Appalachia (MMcf/d)	150	300
Underground Storage (Bcf)	22	44
Appalachia to East Coast (MMcf/d)	407	814
Cost of Gas in Appalachia = 35¢/Mcf x V x 365 days (MM$/yr)	19.25	38.5
Cost of underground storage (13% capital recovery (MM$/yr)	4.55	9.1
Cost of pipeline to East Coast = 12¢/Mcf x V x 365 (MM$/yr)	6.60	13.20
Subtotal	30.40	60.80
Value of gas (¢/Mcf)	55.27	55.27
LNG		
Imports (MMcf/d)	150	300
Extra LNG storage (Bcf of gas)	18	37
Extra Regasification (MMcf/d)	257	514
Cost of extra storage (MM$/yr) = 1.70 x Bcf x .15	4.59	9.47
Cost of Extra Regasification (MM$/yr)	0.20	0.40
Total (MM$/yr)	4.79	9.87
Value (¢/Mcf)	8.7	9.0
Base Load (Table 2)	53.2	46.5
Value of Gasified LNG (¢/Mcf)	61.9	55.5

at both 150 and 300 MMcf/d; the value to the pipeline company is 55.3¢/Mcf in both cases.

The LNG storage srequirement has been partially credited with that part of the terminal storage which can be assigned for seasonal load balancing, namely 4 Bcf for 150 MMcf/d and 7 Bcf for 300 MMcf/d. Extra storage has been costed at $1.70/Mcf of equivalent gas, i.e. about $6.00 per barrel, a value corresponding to the maximum storage size currently considered for above-ground tanks. Extra regasification equipment is also taken into account. The new value of the cheapest (Caribbean) gasified LNG is still 6.6¢/Mcf higher than domestic gas in the 150 MMcf/d case, and about on par in the 300 MMcf/d case.

(2) Needle Peakshaving:

Figure 5 illustrates a peak load over 15 days in a normal year on the East Coast. The solid rectangle represents base load, and the dashed line indicates the extra amount needed to compensate for the peak if spread over the entire year. Table 4 compares two methods of needle peakshaving: 1) piping gas to the consuming centers, liquefying it, storing it, and regasifying it when needed; and 2) importing additional LNG, storing it, and regasifying it. Two rates of delivery are again considered.

For method (1) we have assumed a design storage capacity of 10 billion cubic feet, corresponding to a typical ten days of regasification at the rate of 1 Bcf per day, and a design liquefaction rate of 50 MMcf/d for 200 days although only 7 Bcf/yr would be liquefied and stored on the average. Spread over deliveries of 150 and 300 MMcf/d

TABLE 4

NEEDLE PEAKSHAVING

(V = 150 or 300 MMcf/d)

PIPELINE LNG		
Gulf to NY, Philadelphia (MMcf/d) 50 MMcf/d liquefaction 10 Bcf storage 1000 MMcf/d regasification	150	300
Cost of Domestic Gas in NY (MM\$/yr) = $V \times 365 \times 0.40 \times 10^{-3}$	22	44
Cost of Liquefying 7 Bcf/yr (MM\$/yr)	2.8	2.8
Cost of Storage (MM\$/yr) = (10 x 1.70) x 0.15	2.55	2.55
Cost of Regasifying 1 Bcf/d (\$MM/yr)	0.6	0.6
Subtotal	27.95	49.95
Average Value of Domestic Gas (¢/Mcf)	50.81	45.41
IMPORTED LNG		
Imports (MMcf/d)	150	300
Extra LNG Storage (Bcf of gas)	9	8
Extra Regasification (MMcf/d)	1000	1000
Cost of Storage (MM\$/yr)	2.3	2.04
Cost of Regasification (MM\$/yr)	0.6	0.6
Subtotal	2.9	2.64
Value (¢/Mcf)	5.27	2.4
Base Load (Table 2)	53.2	46.5
Value of Gasified LNG (¢/Mcf)	58.5	48.9

respectively--i.e., 7.7 and 3.5¢/Mcf more than domestic gas. Under favorable operating experience LNG peakshaving plants could reach twice the volumes implied by present announcements, plans, and construction (1.3 Bcf/d regasification). (14)

ECONOMIC FEASIBILITY

The potential sources of LNG imports to the U.S. East Coast have been identified as Venezuela and the Caribbean for limited amounts, and North Africa for larger volumes; the economics for individual project sizes of 100 to 600 MMcf/d are shown in Figure 3. Even the closest source cannot compete with 40¢/Mcf domestic gas, as shown in the first line of Table 5.

Regulations and air pollution control raise market prices by 3¢/Mcf (600 MMcf/d) to 5¢/Mcf (300 MMcf/d). Projects larger than 600 MMcf/d could probably be fully competitive, but at the risk of market and supply disruption.

We developed conceptual credits on 150 and 300 MMcf/d projects for peakshaving, both in seasonal load balancing and needle LNG peakshaving (Table 5). These point to a potentially competitive 300 MMcf/d project, with load balancing storage of 22 Bcf.

Projects of these smaller sizes also fit well with the market that can be displaced from coal among the intermediate industrial users following the enforcement of air pollution control regulations. These users participate somewhat in the seasonal balancing pattern, and could probably combine seasonal peakshaving credit with air pollution credit, to a total of about 10¢/Mcf versus market price at 40¢/Mcf.

TABLE 5

CONCEPTUAL CREDITS FOR LNG IMPORTS ON THE USNH

(¢/Mcf)

	Delivery Rate (MMcf/d)		
	150	300	600
1. Base load LNG cost differential above 40¢/Mcf domestic gas	13.2	6.5	3.3
2. Seasonal load balancing (Table 3):			
cost differential above domestic gas	6.6	0.2	
Credit vs. base load LNG	6.6	6.3	
3. Needle peakshaving (Table 4):			
cost differential above domestic gas	7.7	3.5	
Credit vs. base load LNG	5.5	3.0	
2 + 3 Seasonal and needle peakshaving			
Credit for	12.1	9.3	

Needle peak loads are primarily created by domestic and commercial heating demands. This market should not be much affected by the new air pollution regulations but would be well suited to a combination of LNG load balancing and peakshaving, bringing again credits of about 10¢/Mcf versus pipeline domestic gas.

Such credits are slightly below what would be necessary to justify a 150 MMcf/d project, but they make Caribbean LNG projects in the 300 MMcf/d range potentially attractive. Their exact value should be assessed in specific cases, since our aggregate model gives only indicative numbers. For example, high construction and land cost could push LNG storage to $10/bbl and reduce load balancing credits to a fraction of a cent per Mcf.

CONCLUSION

Since gas transmission and distribution companies are unsure of future demand growth, they are understandably concerned about domestic gas availability. Nevertheless, the determination of the Federal Power Commission to assert its jurisdiction over LNG imports and to ensure that it should be competitive on economic grounds justifies a careful assessment of every credit which could improve the economics of gasified imported LNG.

The U.S. East Coast alone could support early LNG imports. Our investigation has shown that Venezuela and the Caribbean have currently a 10 to 15¢/Mcf advantage over North African sources for this market, but have smaller potential for natural gas availability. We doubt that special by-product credits could enable exporters or importers to close this gap.

Although economies of scale favor the larger projects, the limitation of risk in smaller projects makes them attractive. Volumes in the order of 300 MMcf/d may be viable economically, if one is able to take advantage of some 10¢/Mcf credit stemming from:

- limited market displacement caused by air pollution control, at the expense of coal rather than fuel oil (also applicable to domestic gas)
- seasonal load balancing
- needle peakshaving

Participation of major pipeline companies is expected. They have been preparing for such opportunities through research, study, and contacts, although generally they have been considering larger project sizes and have yet to set up a project. Shifting the target to a smaller project along the lines suggested above provides an alternate approach. Distribution utilities will find it harder to participate, unless they own conveniently located LNG peakshaving storage with suitable land expansion capability and deepwater access.

In the best case, however, it takes about 2 to 3 years to bring the necessary installations and equipment into service from the contracting date. With an additional two years for studies and negotiations, steady state LNG imports in the United States would not come until 1974. High expectations or early success in the current offshore prospecting activity would undoubtedly cause further delay.

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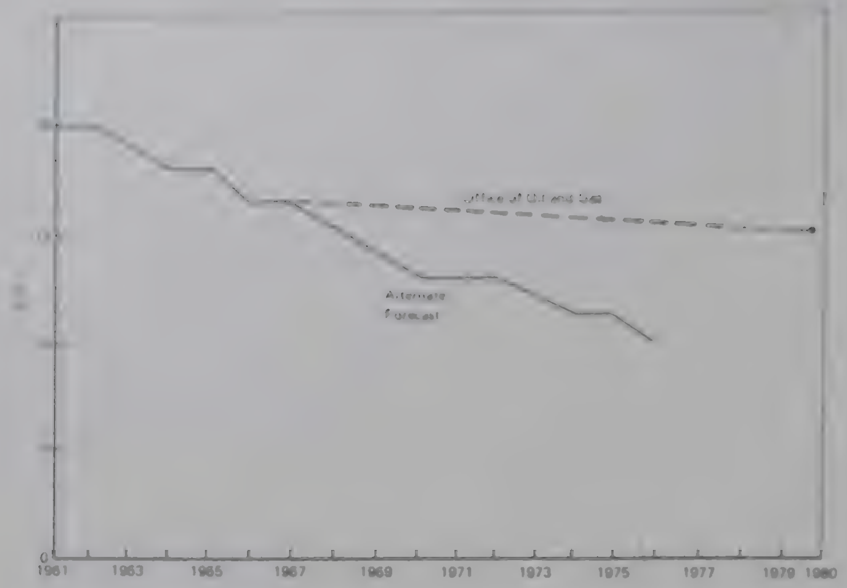

FIGURE 1 CONTIGUOUS U.S. RESERVE–TO–PRODUCTION RATIO FOR NATURAL GAS

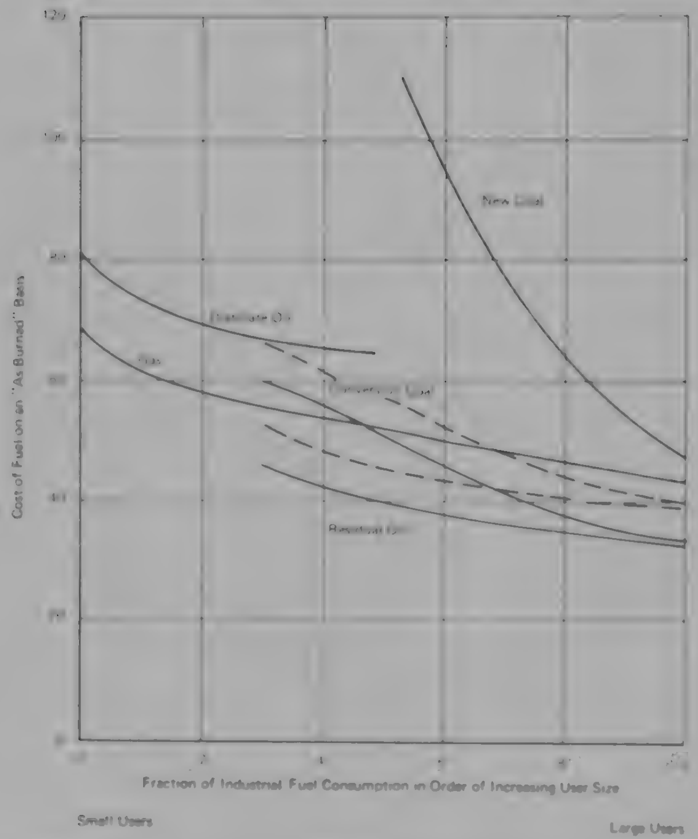

FIGURE 2 U.S. EAST COAST COMPETITIVE INDUSTRIAL FUEL ECONOMICS

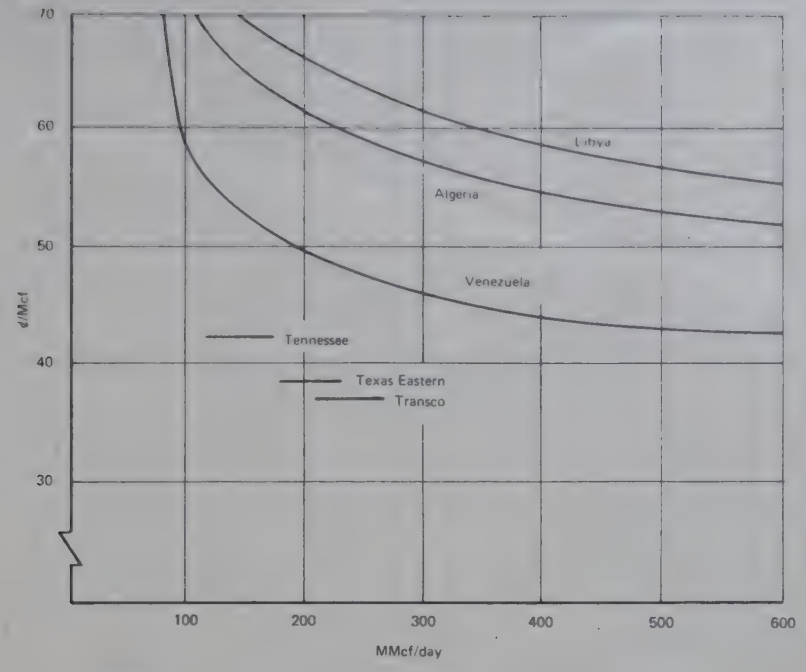

FIGURE 3 GASIFIED TANKER LNG (15% CAPITAL CHARGE) VERSUS DOMESTIC GAS SUPPLY – U.S. EAST COAST

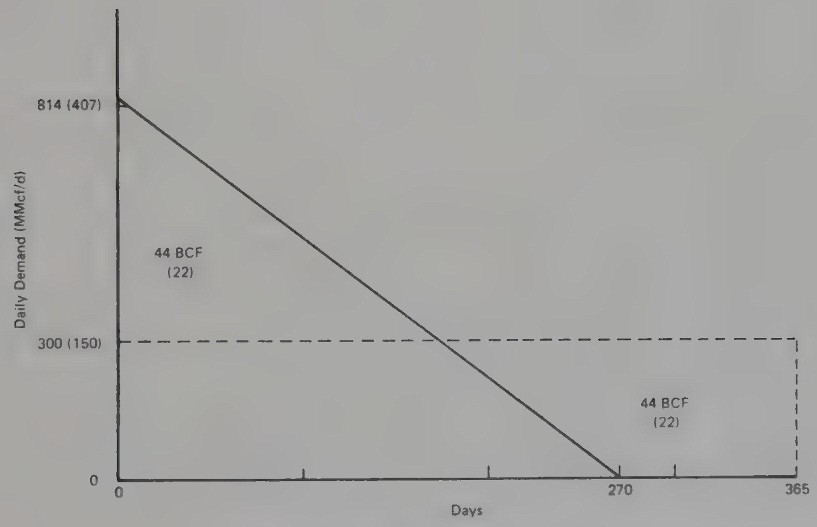

FIGURE 4 SEASONAL LOAD BALANCING PROJECT A

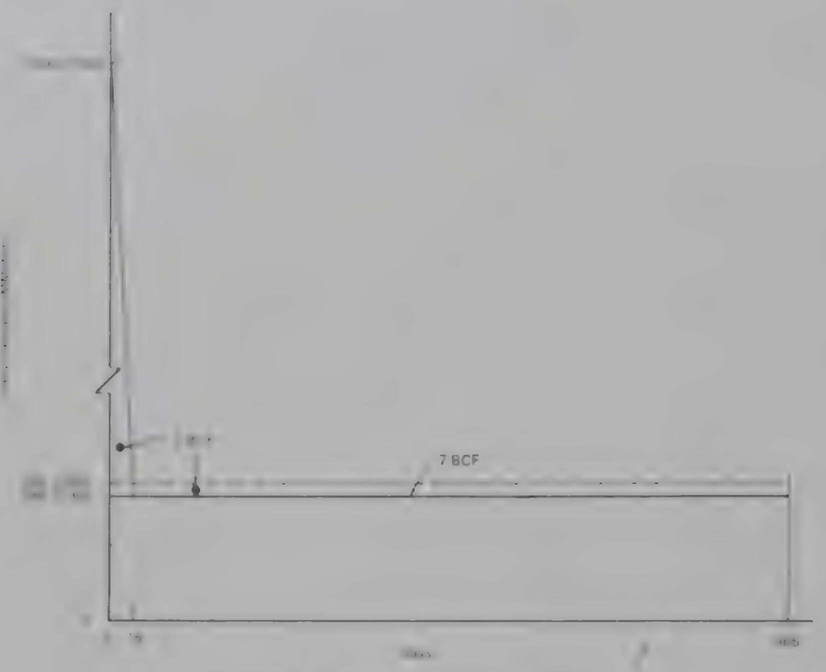

FIGURE 5 NEEDLE PEAK LOAD PROJECT B

THE POTENTIAL OF LIQUEFIED NATURAL GAS AS A SOURCE FOR PETROCHEMICAL FEEDSTOCK

By P.Spencer *

Introduction

There is apprehension on the petrochemical scene as to future supplies of naphtha for ethylene production. Ethylene is the ground floor building block for the petrochemical industry. Japan, the second largest ethylene producer, is already importing 15% of its naphtha requirement and anticipates to triple its production by 1972. The shortage of naphtha in the U.S.A. has been mitigated by the abundance of rich natural gas which has been a source of ethane and propane, their main feedstocks for ethylene production. However, U.S.A. petrochemical producers anticipate this source failing to keep pace with increasing demand, and consequently are turning towards naphtha to correct this unbalance. Looking nearer home, Western European producers expect that with the present oil refining processing structure and that anticipated into the 70's, the naphtha availability should marginally meet demands up to the mid-70's. (1975)

In view of this anticipated world-wide shortage of naphtha, it can be expected that the current interest in processing techniques capable of economically producing ethylene from alternative feedstocks will be accelerated. With this incentive, and the projected increase in world-wide LNG traffic, the potential of exploiting LNG as:

a. A feedstock for ethylene production

* Air Products Limited.

b. A means of reducing operating cost by utilising the refrigeration potential of LNG

has been evaluated, and a technical position developed.

What LNG Feedstocks are Suitable for Petrochemical Production

Since the petrochemical industry is orientating itself more and more towards petrochemical complexes, based on ethylene as opposed to acetylene, only ethylene production and associated by-products will be considered in this paper. The components of LNG which we are interested in for ethylene production are ethane and the heavier hydrocarbons. Hence, any LNG would be suitable for ethylene production which contained a reasonable quantity of ethane and heavier hydrocarbons. An ethane content of 5 mol % is suggested as the lower limit.

Ethylene is made by thermally cracking light paraffinic hydrocarbons in the presence of steam.

The following are typical yields that can be obtained from the individual cracking of the respective components that can be recovered from LNG.

TABLE I

Yields from Thermal Cracking (Optimised for Ethylene)

Weight %

Feedstock	Ethane	Methane	N. Butane	Ethane Feedstock with Ethane Recycle Cracking the Ethane to Extinction
Components:				
Hydrogen	3.6	1.6	0.8	5.5
Methane	4.3	23.5	18.6	3.3
Acetylene	0.3	0.5	0.3	0.5
Ethylene	47.6	32.0	30.0	81.0
Ethane	40.0	5.7	5.4	-
Propylene	1.4	28.3	19.8	4.5
Butadiene	1.7	4.0	18.5	1.8
Aromatic	0.7	2.8	3.0	2.4
Fuel Oil	0.4	1.6	3.6	1.0
Total	100.0	100.0	100.0	100.0

As can be seen, the yield patterns are markedly different for the respective components. This illustrates the significance of the relative composition of the components heavier than methane in evaluating a particular LNG as a potential petrochemical feedstock. Diagram I shows the significance of ethylene as the building block from which so many products are made.

Reduced Operating and Capital Cost by Utilising the Refrigeration Potential of LNG for Ethylene Production

This is where the real potential lies in exploiting LNG as a petrochemical feedstock.

There are two phases involved:-

a. To separate the ethane and heavier hydrocarbons from the LNG.

b. Utilise the excess refrigeration potential in the separation and purification of ethylene within the ethylene producing facility itself.

The primary use now and in the future of LNG will be as a fuel. As the gas load on any grid system is variable both daily as well as seasonal, it is essential that the separated fuel component, namely methane, can be fed into the gas distribution system on a variable flow basis. However, it is just as essential that the feedstock to an ethylene producing facility should be on a steady state basis. This conflicting interest has probably been the main technical restraint to date on exploiting LNG's refrigeration potential for the recovery and purification of ethylene from cracked gases.

Several people have already proposed flowschemes which integrate the separation of ethane and heavier hydrocarbons from LNG with the ethylene recovery and purification section of an ethylene plant using the refrigeration of the LNG to affect the various separations in complex fractionating systems, e.g. Conch International and The Lummus Company. However, their common approach has been to process a constant quantity of LNG, recover the fractions suitable as feedstock for an ethylene producing facility and pass the balance to a pipeline distribution system, as a gas. This approach has one very serious disadvantage. It restricts them to only consider processing that portion of LNG which would amount to the minimum load requirement of a gas distribution system to satisfy the steady state requirement of an ethylene purification facility. In effect limiting the exploitation of LNG as a feedstock to 30% or even less of that landed. By adopting the following technique, all of the landed LNG could be exploited for its petrochemical feedstock potential and a higher proportion of the potential refrigeration than previously possible.

In essence, 100% of the landed LNG would be processed to separate the ethane and heavier hydrocarbons from the methane. The minimum load requirement for the gas distribution system would be satisfied with separated methane and the balance of the methane would be re-liquefied and stored to meet peak load requirements (refer to diagram II).

Taking an Algerian LNG as a basis for illustration, having the following composition:

	Wt. %
Nitrogen	0.5
Methane	74.4
Ethane	15.1
Propane	6.2
Butanes	3.4
Pentanes +	0.4
	100.0

it was found there was sufficient refrigeration at the appropriate levels to affect the separation of ethane and heavier hydrocarbons from the LNG. Also, sufficient to satisfy the refrigeration requirement for the ethylene recovery and purification facility of an ethylene plant specifically process designed to be compatible with the available refrigeration levels, except for one or two minor services.

For an LNG facility landing 2.85 MMnm^3/day (100 MMSCF/day) of Algerian gas, a 84,000 ton/year ethylene facility could be provided taking only ethane as feedstock at 90% recovery from the LNG. Diagram III shows the mass and energy balance for an integrated Ethane/Ethylene/LNG complex.

The power consumption for a modern conventional naphtha feed ethylene plant is between 1.2 and 1.5 KW/Kg of ethylene produced. This is approximately consumed as follows:-

Cracked feed gas compression	30%
Propylene & Ethylene (refrigeration systems)	70%
	100%

Other power consuming services are negligible in comparison.

The new ICI naphtha feed 450,000 ton/year ethylene plant is reported to require 33,000 bhp. On a straight proration basis, to produce 84,000 ton/year ethylene from LNG as opposed to naphtha would save:-

$$\frac{84,000}{450,000} \times (0.7)\ (33,000)$$

$$= \quad 4,312 \text{ bhp}$$

The saving would not only be realised in utilities alone, but also in capital investment, as no refrigeration compressors would be required and other associated capital equipment.

Economic Justification for the Production of Ethylene from LNG

The potential and feasibility of producing ethylene from LNG has been illustrated. It now remains to show that producing ethylene from a suitable LNG, in limited quantities, is more economical than from naphtha. This is presented in Tables II and III.

Table II - Shows an evaluation of an ethane from LNG recovery plant.

Since an ethane and heavier hydrocarbons from LNG recovery plant could be built as a separate self-contained unit, the economic justification for such a facility has been treated separately. The capital investment and operating cost has been equated against the value of recovered propane and butane as LPG and the pentanes as a petrol blending component, whilst the recovered ethane has been credited with its fuel value only, so that the subsequent cost comparison for the production of ethylene from naphtha as opposed to ethane recovered from LNG will be on a common basis. Naphtha is sold as a commodity whether it is used as:-

a. Feedstock to an ethylene plant

b. Naphtha reforming for gas production

c. Ammonia production

is irrelevant to the retailer.

Table III - Shows, by comparison, that ethylene can be produced cheaper from ethane recovered from LNG than from naphtha.

The main points to note are:-

a. Reduced utility requirement. Lower power, cooling water, etc.

b. Higher labour operating costs. Although the process is simpler and maintenance cost should be less, labour costs per ton of ethylene made will be higher.

TABLE II

ECONOMIC EVALUATION OF ALL ETHANE FROM LNG RECOVERY PLANT.

Basis: 2.85MM NM^3/PAY (100 MM scfd) of Algerian LNG.

Component	Feed Gas Composition Mol %	Feed Gas Composition Wt. %	% Recovered	lb/day Recovered
Nitrogen	0.3	0.5		
Methane	86.5	74.4		
Ethane	9.4	15.1	90	665500
Propane	2.6	6.2	100	303800
Butane	1.1	3.4	100	166600
Pentanes +	0.1	0.4	100	19600
	100.0	100.0		1155500

	Cost, £ per year
Raw Material/Operating Costs	
Feedstock cost (C_2 + only - fuel value @ 3d/therm)	1,032,000
Utilities	20,000
Operating Cost (labour and overheads)	100,000
Sub Total	1,152,000
Fixed Capital Charges/Overheads	
Taxes/Insurance/Maintenance etc. (5% capital investment)	100,000
Sales/Administration/Return on working capital (5% capital investment)	100,000
Sub Total	200,000
Credits	
Revenue from ethane sales (Fuel value @ 3d/therm equivalent £5.7/ton)	592,000
Revenue from propane sales @ £18/ton	855,000
Revenue from butane sales @ £20/ton	520,000
Revenue from C_5 + sales @ £12/ton	40,000
Sub Total	2,007,000

Net cash inflow per year 2,007,000 - 1,352,000

= £655,000/year.

Estimated plant capital cost £2.0 MM

Using discounted cash flow method, True Rate of Return = 28%

TABLE III

COMPARISON OF OVERALL COST FOR THE PRODUCTION OF ETHYLENE

Basis: Costs for a 450,000 T/A ethylene unit (conventional - naphtha)
Costs for a 84,000 T/A ethylene unit (as integrated with an ethane/ LNG recovery plant)

	Cost / ton Ethylene (£)		
Feedstock	Naphtha	Ethane from LNG	
		Optimistic	Pessimistic
Raw Material/Operating Costs			
Net feedstock cost	4.72(i)	3.95(ii)	5.72(iii)
Catalyst and chemicals (Mainly caustic soda)	1.00	0.50	0.50
Utilities	7.56	3.70	3.70
Operating cost (labour and overheads)	0.40	0.70	0.70
Sub Total	13.68	8.85	10.62
Fixed Capital Charges/Overheads			
Taxes/insurance/maintenance (5% cap. invest.) Return on investment (25% cap. invest.)	9.85	7.86	7.86
Sales/Administration/Return on Working Capital (5% cap. invest.)	1.64	1.64	1.64
Sub Total	11.49	9.5	9.5
Credits.			
H.P. steam export		1.50(iv)	
TOTAL £/ton Ethylene	25.17	16.85	20.12

i Naphtha cost £12/ton. Co-products credited at premium values.

ii Ethane cost at fuel value of 3^{d}/therm. Co-products credited at premium values.

iii Ethane cost at fuel value of 3^{d}/therm. Co-products credited at fuel value.

iv Steam credited at 4s. per 1,000 lbs. exported.

c. Lower fixed capital charges. No expensive refrigeration machinery required and associated equipment - the process is simpler. Capital investment has been estimated at 80% of conventional capital cost, which is considered to be a conservative estimate.

Other Advantages of LNG as a Feedstock for Ethylene Production

1. As virtually all current large tonnage transportation of LNG is effected in special cryogenic sea tankers, and will continue to be in the future, the same docking and transport facilities could be utilised for the shipment of ethylene product to consumer areas, in particular to the coastal terminals of the projected ethylene pipeline distribution system, within Europe, as well as the U.K., e.g. ICI ship ethylene by tanker from Wilton - U.K. to Rozenburg - Netherlands.

2. A considerable quantity of low grade heat is generated in an ethylene plant which is of nuisance value. This could be utilised to vaporise LNG

3. By utilising a high proportion of landed LNG for producing ethylene, more favourable contract terms for the utility company should ensure for the purchase of LNG, as larger quantities would be required.

4. Amortisation of capital investment to the utility company would be quicker with respect to the docking and shipping facilities.

Some Disadvantages of LNG as a Feedstock for Ethylene Production

1. Although much smaller capacity production units could be economically justified against the new outsized 500,000 ton/year naphtha feedstock ethylene plants currently being built, the size of unit would be limited by the utility companies requirements.

2. Since the current sources of LNG are restricted to only a few locations, the political security would not be as great as with a naphtha feedstock ethylene plant.

3. If the ethylene producing operating company was not the utility company controlling the LNG, then the operating company would have to carry the additional risk of its dependence on the utility company.

Conclusions

1. The technical feasibility is established using present day technologies developed for the liquefaction of natural gas and those for designing conventional ethylene plants.

2. The economics of producing ethylene from LNG and the logical extension of developing a petro-chemical complex based on LNG looks very lucrative.

3. Dependence on the utility company operating the LNG vaporisation facility from a petro-chemical producers position is not desirable, nor the limited production aspect.

4. If there is a projected price increase for naphtha, and the technology of using other petroleum cuts, such as gas oil, is not developed to be economically competitive, then increased interest will be given to exploiting LNG for ethylene production, particularly if the boom in LNG traffic continues at its present rate.

5. Finally, this new avenue for producing ethylene opens the door to interested newcomers as well as established ethylene producers, particularly utility companies, who are currently discouraged by the new wave of outsized ethylene plants based on naphtha feedstocks which would make a modest entry totally uneconomic following the conventional route.

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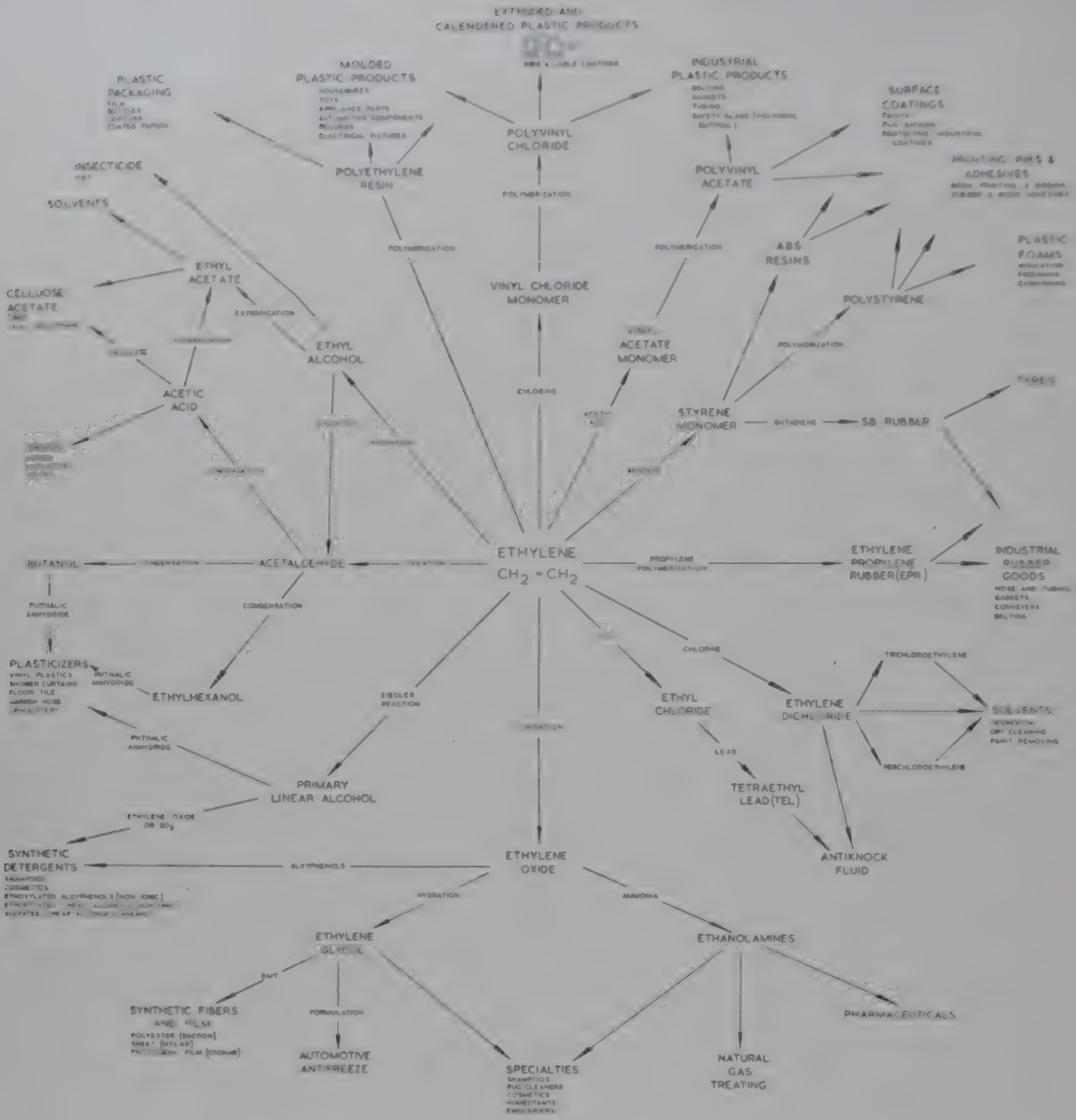
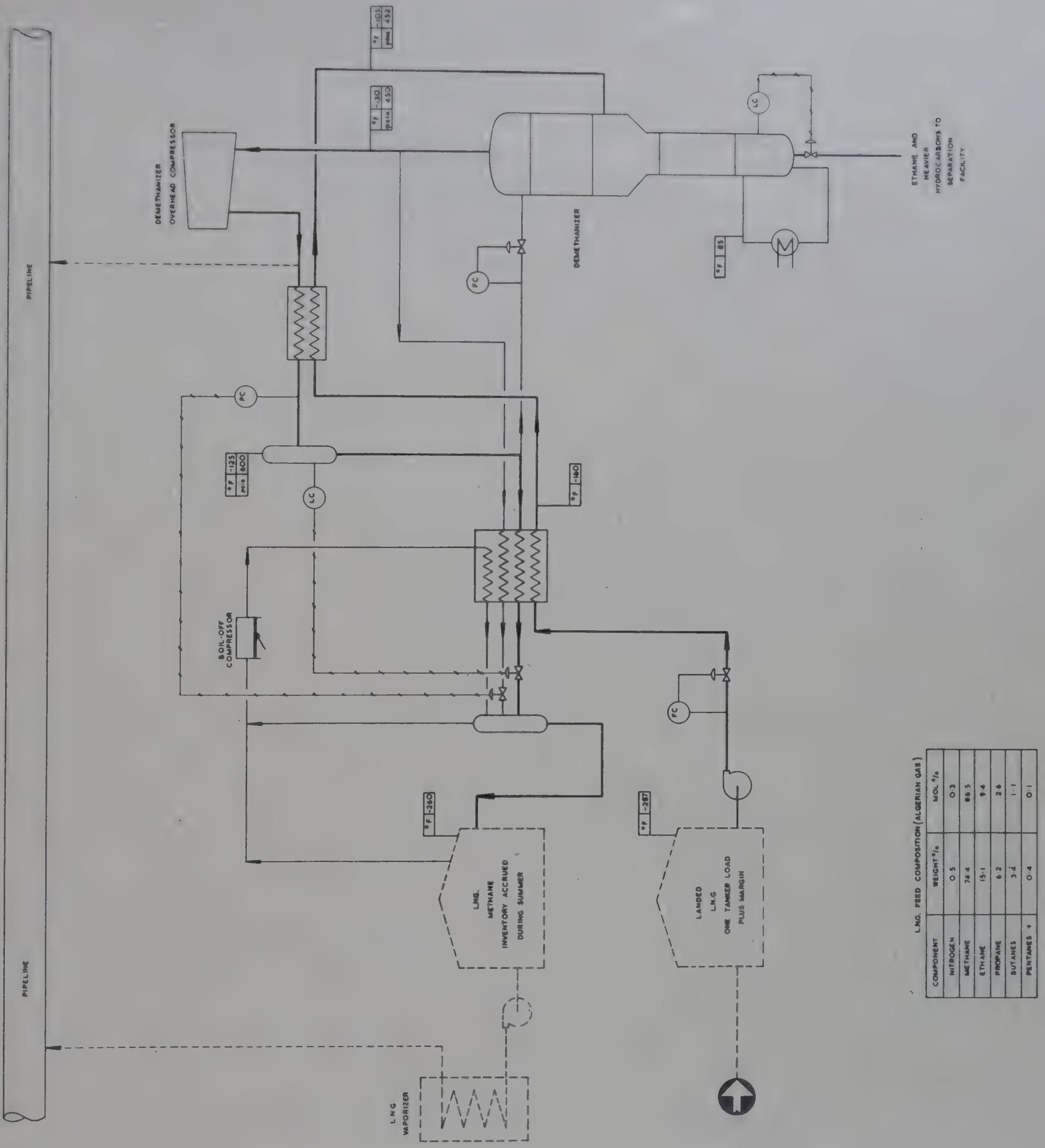

EXTRUDED AND CALENDERED PLASTIC PRODUCTS
WIRE & CABLE COATINGS
MOLDED PLASTIC PRODUCTS
HOUSEWARES
TOYS
APPLIANCE PARTS
AUTOMOTIVE COMPONENTS
CLOSURES
ELECTRICAL FIXTURES
PLASTIC PACKAGING
FILM
BOTTLES
COATED PAPERS
INDUSTRIAL PLASTIC PRODUCTS
GASKETS
TUBING
SAFETY GLASS (POLYVINYL BUTYRAL)
SURFACE COATINGS
PAINTS
PROTECTIVE INDUSTRIAL COATINGS
PRINTING INKS & ADHESIVES
PLASTIC FOAMS
INSECTICIDE
DDT
SOLVENTS
POLYVINYL CHLORIDE
POLYETHYLENE RESIN
POLYVINYL ACETATE
POLYMERIZATION
ABS RESINS
ETHYL ACETATE
CELLUOSE ACETATE
ESTERIFICATION
VINYL CHLORIDE MONOMER
POLYSTYRENE
ETHYL ALCOHOL
VINYL ACETATE MONOMER
ACETIC ACID
CHLORINE
TYRES
STYRENE MONOMER
BUTADIENE
SB RUBBER
HYDRATION
BENZENE
BUTANOL
CONDENSATION
ACETALDEHYDE
ETHYLENE
$CH_2=CH_2$
PROPYLENE POLYMERIZATION
ETHYLENE PROPYLENE RUBBER(EPR)
INDUSTRIAL RUBBER GOODS
HOSE AND TUBING
GASKETS
CONVEYERS
BELTING
PHTHALIC ANHYDRIDE
CONDENSATION
PLASTICIZERS
VINYL PLASTICS
SHOWER CURTAINS
FLOOR TILE
GARDEN HOSE
UPHOLSTERY
PHTHALIC ANHYDRIDE
ETHYLHEXANOL
ZIEGLER REACTION
ETHYL CHLORIDE
CHLORINE
ETHYLENE DICHLORIDE
TRICHLOROETHYLENE
SOLVENTS
DRY CLEANING
PAINT REMOVING
PERCHLOROETHYLENE
PHTHALIC ANHYDRIDE
LEAD
PRIMARY LINEAR ALCOHOL
TETRAETHYL LEAD(TEL)
ETHYLENE OXIDE OR SO_3
ANTIKNOCK FLUID
SYNTHETIC DETERGENTS
SHAMPOOS
COSMETICS
ETHOXYLATED ALKYPHENOLS (NON IONIC)
ALKYPHENOLS
ETHYLENE OXIDE
HYDRATION
AMMONIA
ETHYLENE GLYCOL
ETHANOLAMINES
DMT
SYNTHETIC FIBERS AND FILM
POLYESTER (DACRON)
SHEET (MYLAR)
FILM (CRONAR)
FORMULATION
AUTOMOTIVE ANTIFREEZE
SPECIALTIES
SHAMPOOS
RUG CLEANERS
COSMETICS
HUMECTANTS
EMULSIFIERS
NATURAL GAS TREATING
PHARMACEUTICALS

Fig.1. Ethylene - the ground floor building block for petrochemicals


DEMETHANIZER OVERHEAD COMPRESSOR
°F -105 psia 452
°F -30 psia 450
LC
ETHANE AND HEAVIER HYDROCARBONS TO SEPARATION FACILITY
DEMETHANIZER
°F 85
PC
PIPELINE
PC
°F -125 psia 600
LC
°F -160
BOIL-OFF COMPRESSOR
FC
°F -260
L.N.G. METHANE INVENTORY ACCRUED DURING SUMMER
°F -257
LANDED L.N.G ONE TANKER LOAD PLUS MARGIN
PIPELINE
L N G VAPORIZER

L.N.G. FEED COMPOSITION (ALGERIAN GAS)

COMPONENT	WEIGHT %	MOL %
NITROGEN	0.5	0.3
METHANE	74.4	86.5
ETHANE	15.1	9.4
PROPANE	6.2	2.6
BUTANES	3.4	1.1
PENTANES +	0.4	0.1

Fig.2. Flow scheme for ethane and heavier hydrocarbons recovery from LNG plant

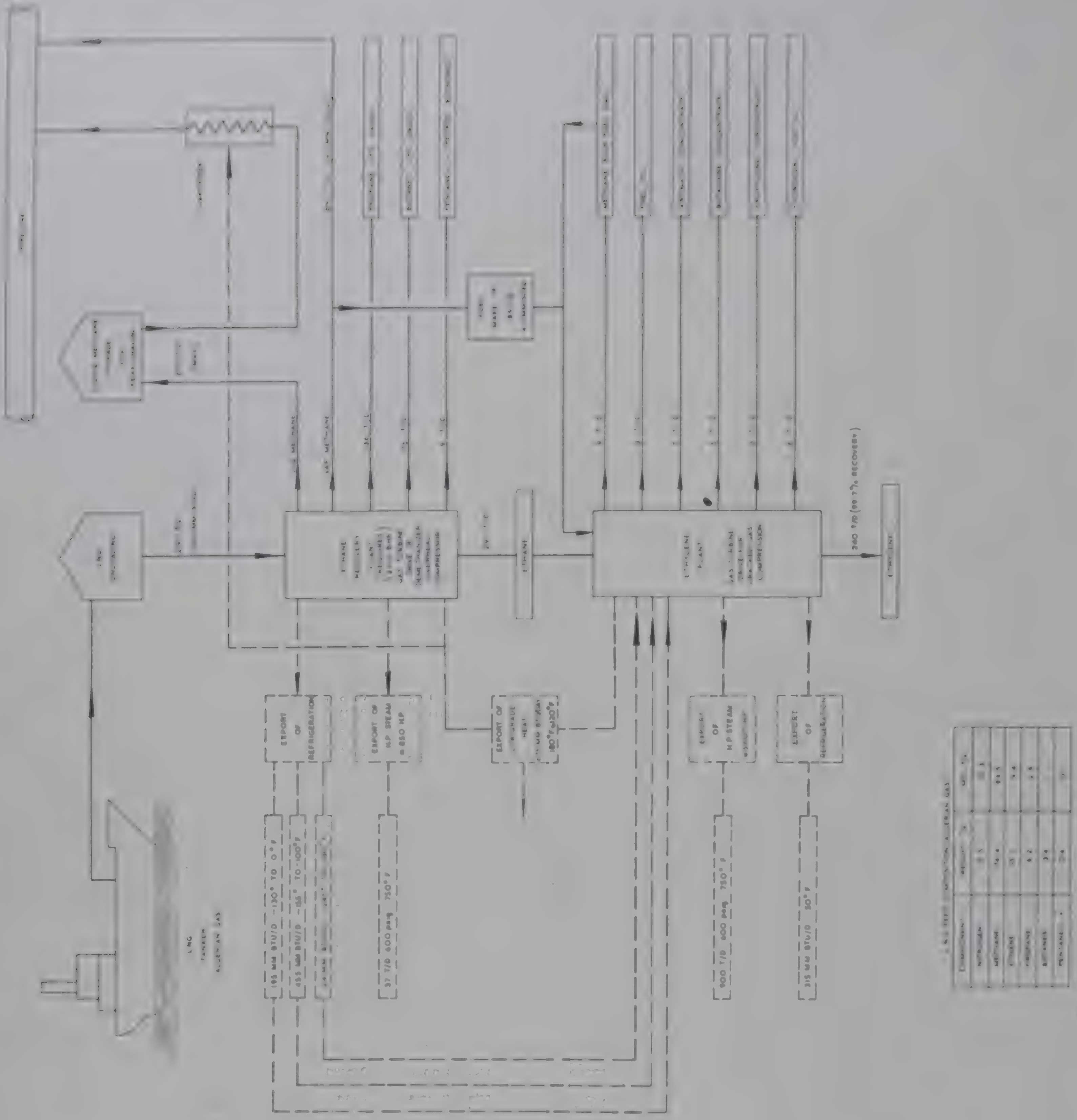

LNG TANKER ALGERIAN GAS
ETHANE
ETHYLENE
240 T/D (99.7% RECOVERY)
EXPORT OF REFRIGERATION
EXPORT OF H.P. STEAM
EXPORT OF H.P. STEAM
EXPORT OF REFRIGERATION
195 MM BTU/D -130° TO 0° F
57 T/D 600 psig 750° F
900 T/D 600 psig 750° F
315 MM BTU/D 50° F

Fig.3. Material and energy balance for ethane/ethylene/LNG complex

NEW APPLICATIONS FOR LNG

by

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President, LNG SERVICES
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Because LNG has so many new and potential applications as a fuel, this paper will discuss some of the more immediate markets for LNG including various transportation applications, standby fuels for community and industry, as well as future innovations.

Transportation applications include: over-the-road equipment such as automobiles, taxicabs, buses, trucks; off-the-road units such as trucks, scrapers, and other earth-moving equipment; railroad locomotives for rapid transit, marine transportation systems, and aircraft and helicopter fuel.

The portability of LNG lends itself to base load supply to outlying communities and temporary fuel for service line interruptions. LNG fueled standby electric power generation, desalinization of water, and magnetohydrodynamic (MHD) generation of electricity are also discussed.

Applications for Over-the-Road Equipment

Liquefied natural gas (LNG), a supercold, high-energy fuel, is being considered principally because it is a clean-burning fuel with a smokeless exhaust, it lowers maintenance costs, and it is priced lower than other fuels in many areas.

Its hot performance in the fight against automotive smog has been clearly demonstrated by the San Diego Gas and Electric Company. For the past two years, they have been operating a 1967 standard 6-cylinder Dodge pickup truck and a 1967 6-cylinder Valiant on LNG supplied from their Chula Vista, California LNG plant.

In their latest test, the LNG vehicles emitted less than half as much smog-producing pollutants as the average gasoline-fueled cars equipped with smog control devices. Although the LNG cars were also equipped with smog control devices, tests with the device disconnected showed the same low percentage of smog-producing emissions.

Since June 6, 1968, two 1968-model passenger cars were used in 50,000-mile tests to determine what changes occur in exhaust emissions as a result of engine wear.

The California Air Resources Board conducted exhaust emission tests before the start of the 50,000 mile test. Results of exhaust emission tests on unburned hydrocarbons and carbon monoxide is shown in Figures 1 and 2. Present California exhaust standards for air pollution control allow 275 parts per million hydrocarbons and 1.5 percent carbon monoxide.

Test vehicle conversion to LNG included installation of an insulated 21 gallon fuel tank to maintain the extremely cold temperature required to keep the gas liquid, a coil of copper tubing to act as a vaporizer to change the liquid back into a gas, a gas regulator, and a natural gas carburetor. In addition, the engine cylinder head was milled to increase the compression ratio. Because LNG is a slower burning fuel than gasoline, a new camshaft was installed to keep the exhaust valve closed longer permitting more complete combustion. The ignition timing also was altered slightly.

When the vehicle is in operation, LNG flows from the insulated tank into a copper coil. The air surrounding the coil adds heat to the LNG, turning it back into a gas. The regulator controls the flow of natural gas into the carburetor where it is mixed with the proper proportion of air to make it combustible. This mixture is then fed into the engine.

"Dual-fuel" Automobiles

LNG SERVICES is operating two "dual-fuel" system automobiles which operate alternately on gasoline and natural gas. They are a 1969 Pontiac and a 1969 Oldsmobile. The system was designed to permit motorists to burn LNG while driving at uneven speeds in traffic, when most exhaust pollution occurs, and to switch to gasoline for high speed travel.

The Pacific Lighting Corporation of Los Angeles is operating three automobiles on compressed natural gas and one 1968 Ford Ranchero V-8 pickup truck using LNG as shown in Figure 3. The LNG is stored at -260°F in a 60-gallon insulated vacuum tank located in the truck bed. This supply is enough for 750 miles of travel. Figure 4 illustrates the engine which is similar to the compressed natural gas system except for the addition of the vaporizer equipment by the radiator.

Test results show that a vehicle operating on natural gas had carbon monoxide emissions reduced 90 percent from 1.3 percent in gasoline-operated vehicles to 0.13 percent in the Ford. Hydrocarbons were 88 parts per million of air versus 221 for gasoline, and oxides of nitrogen were reduced from 1,070 to 554 parts per million of air.

Operating costs have been found to be substantially lower for natural gas-powered vehicles with fuel costs almost halved. One fleet of cars using natural gas averaged 14.4 miles per 100 cubic feet, costing about 7 1/2 cents, not including 11 cents road tax (79.5 cubic feet of gas equals 1 U. S. liquid gallon). By comparison, the gasoline-driven cars averaged 13.6 miles per gallon of premium-grade gasoline, costing 23 cents, not including road tax.

Lower oil and maintenance costs have also been found. The clean burning nature of natural gas does not create sludge and dilute the oil, significantly reducing the number of oil changes needed. Since it does not act as a solvent, it does not remove oil from the cylinder walls. It also leaves spark plugs carbon free for better performance and longer life and provides smoother combustion performance since it is completely vaporized when it reaches the carburetor.

LNG-fueled Buses and Taxicabs

Although LNG SERVICES is working with several municipalities on test programs to evaluate LNG fueled buses and taxicabs, it is too early to report any results from these test programs.

Currently one evaluation is being conducted with the Yellow Cab Company of Pittsburgh, Pennsylvania. The Cab Company has furnished twelve 1969 vehicles for this experiment. Six of the units are operating on gasoline and six have been converted to operate on LNG. All maintenance and fueling cost records for both fuels are being kept in detail. (See Figure 5.)

All twelve cabs operate 24 hours per day, seven days a week. The duration of the test will be for a minimum of six months, or at least 50,000 miles per cab under all types of weather conditions. Pittsburgh's terrain is quite hilly with high density traffic in the downtown area as well as new high speed freeways to the suburban districts. This provides a representative set of operating conditions on which to base our test results.

The taxicabs are Checker Manufacturing Company units with 6-cylinder, in-line Chevrolet engines (250 cubic inch displacement and 155 horsepower). The engines have been modified to varying degrees to study the differences in maintenance cost and driver reaction to determine what engine modifications are optimum for good operation. Dynamometer tests were conducted both before and after conversion.

Comparative checks of engine exhausts are being made after each four weeks of operation on both types of fuel. The results are being compared with both existing and proposed Federal and State Emission Requirements.

LNG SERVICES plans to publish the results of this experiment detailing driver reaction, comparative maintenance costs, exhaust emission results and fuel handling costs.

LNG-fueled Racing Car

Although racing cars will not be a big market for LNG, they are of interest. Reaction Dynamics, Inc. is working on a program to set three new land speed records: 1) to break the present record, 2) to exceed the velocity of sound on the ground, and 3) to exceed 1,000 mph.

Reaction Dynamics is using an experimental liquid hydrogen peroxide, monopropellant rocket with LNG fuel, and LNG SERVICES is serving as a consultant on LNG handling and storage. Running tests will be conducted at various speeds on drag strips and at the Bonneville Salt Flats during 1969.

Applications for Off-the-road Equipment

The growing acceptance of gas turbine engines will add to the demand for LNG as fuel, since these engines can use vaporized LNG without modifying the engine system. Large, off-the-road vehicles such as those used by United States Steel Corporation in their open pit mining in Minnesota are one potential market.

Currently, United States Steel is conducting a one-year test to compare operating and maintenance costs of two LNG-fueled trucks with two identical diesel fueled vehicles, to determine savings in engine maintenance with LNG fuel. Northern Natural Gas Company is operating an LNG liquefaction plant capable of producing about 600 gallons (48 Mcf) of LNG per day. It is located 1 1/2 miles east of Buhl, Minnesota and 3 miles from the truck-fueling depot. United States Steel transports LNG from the 1500-gallon LNG tank to this fuel depot in a 1050-gallon LNG tank truck. If the test proves successful, United States Steel may convert more of its off-the-road equipment to LNG fuel. The fleet includes diesel trucks, locomotives and bulldozers.

As larger engines are required, ranging from 700 to 1000-hp the gas turbine has become a competitive source of power. Solar Division of International Harvester has accumulated more than 6,000 hours of experience with a 1100-hp Saturn industrial gas turbine in a 100-ton Unit Rig Lectra Haul M-100 at Anaconda's open pit copper mine at Butte, Montana. The LP-gas turbine provides shaft power to a dc generator which, in turn, supplies power to a pair of General Electric traction motors in the drive wheels.

At Butte, the turbine-driven trucks climb 8 to 10 percent grades fully loaded at greater speeds than conventional trucks, and engine maintenance costs have been slashed by about 50 percent. Other Lectra Hauls powered by 1100-hp Saturn gas turbines are operating successfully at Kennecott's Bingham Canyon open pit in Utah and at Kaiser Steel's Eagle Mountain pit in California.

Solar's off-highway turbine engine experience indicates that operators may expect a total operating engine life of about 35,000 hours, and an interval between major overhauls of 8,000 to 12,000 hours.

The Proposed LNG Turbine-Powered Truck

With this operating experience, we believe that turbine-driven engines powered by LNG can offer impressive advantages to open pit truck operators. The proposed 160-ton ore hauler incorporates a 1100-hp gas turbine which drives an electric generator and is propelled by electric motors in four drive wheels. This rig is currently being evaluated at an open pit mine in the United States.

To install this system a standard gaseous fuel system replaces the diesel fuel controls system, a 600-gallon LNG storage tank with a heat gain of less than 2 percent boiloff per day is mounted in the standard fuel storage area, and a heat exchanger is added to convert or vaporize the LNG.

Projected LNG cost estimates to operate 50 trucks are as follows:

1. The basic gas cost is $0.32 per 1,000 cubic feet, or 2.68¢ per gallon.
2. Equipment to liquefy the gas and provide adequate storage consists of a liquefaction plant, storage tanks, 50 fuel systems, and service department. This cost would be $2 million or $0.0162 per gallon, if applied over a five-year period.

3. The operating cost, including utilities and personnel, will be about 10¢ per 1,000 cubic feet, or $0.00836 per gallon.

4. The LNG cost is therefore:

Gas cost	$0.0268
Plant amortization cost	0.0162
Plant operating cost	0.00836
	$0.05136 per gallon

5. Total savings per hour would be 40 gallons per hour x $0.010 - 61.6 gallons per hour x $0.05136 = $0.85 per hour.

6. A 50-truck fleet powered by LNG rather than diesel fuel would annually save: $0.85 x 50 x 22 hours per day x 364 days per year = $341,000 per year.

In addition to using LNG for off-the-road equipment in the mining industry, LNG may be used to supply the entire power requirements for large areas such as the Quebec-Labrador iron mining field in Canada. By installing large supply plants at docks near Port Cartier or at Seven Islands, not only trucks but diesel electric locomotives and open pit earth-moving fleets can be converted to LNG operation to justify the investment in LNG liquefaction plants and cryogenic trucks. Also being considered is using an alternator to replace the generator, as has been done in some prototype diesel-electric locomotives in Europe.

Other Applications in Transportation

Other markets for LNG fuel include rapid transit trains, marine transportation, aircraft and helicopters. Although these markets are not as immediately apparent as automotive or truck applications, they could develop into large future markets for LNG.

Rapid Transit Trains

Several railroads are already testing gas turbine trains, for example, the United Aircraft Corporation is building two, three-car, TurboTrains - turbine-powered, streamlined, and capable of 160 miles per hour, for the New Haven Railroad's New York to Boston run. Service is planned for 1969, with the railroad predicting a one-hour cut in the three hour and 15 minute run. One TurboTrain tested out at 170.8 miles per hour on the Penn Central test tract. The Canadian National Railway will use the turbine trains on its Montreal-Toronto runs, cutting one hour from its present four-hour schedule.

With minor modifications, these engines and others which will operate on a kerosene-based fuel can be adapted to using vaporized LNG as a fuel. This modification can be particularly economical when operating rapid transit trains or high-speed, non-stop transcontinental express freight or passenger trains. Alternate crews and sleeping quarters could be easily incorporated into such train system designs. LNG total energy systems can also be used for all power needs including air conditioning, lighting, and kitchen equipment.

Marine Transportation

In view of the growing acceptance of gas turbines for ship propulsion, the marine field is another potential market for LNG. The United States has accepted five turbine coast guard cutters, four more are under construction and bids for six more are expected soon. Until the application of LNG fuel, turbines required jet fuel, making fuel costs almost double over conventional diesels which used bunker fuel. With the economy of LNG fuel as well as an increase of ten percent cargo space, gas turbine propulsion is being favorably considered.

One specific application would be to high-speed commercial fishing vessels in which the LNG system could be used for total power needs including refrigeration of fish. Another application is high-speed crew boats for servicing off-shore drilling platforms.

LNG as Aircraft Fuel

Another market for LNG has been its highly publicized potential for use as a fuel for second generation supersonic transports. Some predict that the SST program will require gas volumes of about five percent of United States natural gas production by 1987.

Although LNG offers some design difficulties, the concensus seems to be that the advantages, particularly the lower cost of LNG, outweigh the disadvantages which are not insurmountable.

When a liquefaction facility is set up adjacent to an airport to handle LNG demands, the facility could also serve as an LNG peak shaving (or standby fuel) facility for utility service.

Recent studies show that direct operating cost of LNG can be 25 to 35 percent less than JP fuel. In this case, LNG was produced at a liquefaction facility located at the airport, similar to peak shaving plants now in operation.

To put these percentages in more meaningful terms, NASA has estimated that one Mach 3 SST can make three 4000-mile trips each day carrying 200 passengers. A fleet of 500 SST's could conceivably furnish some 400 billion seat-miles per year. Therefore, a reduction of only 0.1 cent per mile saves the airlines $400 million in a year.

Helicopters, VTOL and STOL aircraft present still another application for LNG as an aircraft fuel. Because LNG has lower density than kerosine, its lighter weight will permit helicopters to carry heavier payloads with less fuel weight for the same flying time.

Portability of LNG

Certainly high on the list of advantages for LNG is the ability to easily "move" it around the nation and the world. As the LNG industry grows, increasing need for various LNG transportation systems will be needed to move the product to satellite peak shaving facilities or to other end use customers. LNG has already been transported by truck, tank trailer and ship. Special LNG railroad tank cars have been manufactured, and LNG barges are under consideration for several locations.

Base Load LNG to Outlying Communities

From its LNG plant in Richmond, B. C., Cryogenic Enterprises, Ltd. transports LNG in two tank trucks (see figure 6) to the community of Squamish where it is stored for later regasification and distribution. This is the first known application of its kind in the western world. It is a combination of the satellite peak shaving principle and the concept of a base load LNG plant servicing an outlying community that cannot economically justify pipelined natural gas.

Another noteworthy example is the regularly-scheduled LNG service by San Diego Gas and Electric Company to the Borrego Road Runner Club, a 144-unit mobile home park at Borrego Springs, California. LNG is stored at the park in a 7,750 gallon cryogenic storage tank and is used to meet gas requirements of residential customers and to fuel 116 gas street lights. This is San Diego Gas and Electric Company's second LNG customer. The first was a United States Marine Corps training center located in a remote area of Camp Pendleton, California which has received LNG by truck since August, 1967.

Northern States Power plans to build LNG plants in Eau Claire and La Crosse, Wisconsin, in a piggy-back type of operation which is expected to receive increasing approval from smaller distribution companies. From the larger centrally located liquefaction plants, the liquefied product will be transported to aboveground storage facilities in the local area. The primary plant will be located in Eau Claire and will supply LNG to the La Crosse facility by highway cryogenic transport trailer.

LNG for Service Line Interruptions

Another potential use is vaporizing LNG to supply customers whose regular pipeline service would otherwise be interrupted.

To prove the feasibility of the concept San Diego Gas and Electric carried LNG in a 600-gallon cryogenic portable tank from the Chula Vista plant to Poway, California.

During the 3 1/2 hour test, the tank and its vaporization equipment supplied 1700 residential and commercial customers with 370 gallons of LNG (or 30.8 Mcf.), while a mainline valve was installed in the 16-inch pipeline that supplies the area. This project proved the feasibility of using existing cryogenic hardware with portable LNG to serve gas distribution areas cut off from normal gas supply.

LNG-fueled Standby Electric Power Generation

Another use for LNG is in standby electric power generation. Solar Division of International Harvester Company with the San Diego Gas and Electric Company, has conducted a test of an 800 kw emergency power van operating on LNG fuel. The objectives of the test were to: 1) Investigate the following modes including starting, acceleration, rated loads, transient response. and system stability; 2) Investigate the compatibility of present natural gas hardware; 3) Determine amount of inlet air cooling and attendant problems.

The results showed the feasibility of operating a standard Saturn turbine (equipped for natural gas fuel) on LNG. There was a pickup of 120 horsepower from lowering the inlet air temperature by passing the LNG through an ambient air LNG vaporizer. The vaporizer and ducting were not optimized for the application. Better results would be anticipated had an optimized vaporizer and ducting design been utilized.

Other Industrial Applications

Two final areas for use of LNG in the industrial area that require additional investigation would be its use in magnetohydrodynamic (MHD) generation of electricity and desalinization of water.

In MHD generation of electricity, still in the experimental stage, LNG has been considered both as a fuel for the generator, to produce oxygen for the combustion system and to cool the magnet to increase conductivity.

In desalinization, LNG offers a more than ample heat sink that could be used for producing portable water by freezing. The cost of production of water by this method could be lower than conventional freezing methods, but higher than the current marketable price for water in most areas of the United States.

The Future of LNG

Any one of these potential applications for LNG as a fuel for ground, air, or sea transportation can increase the market for this supercold liquid by several times over what is now predicted for peak shaving purposes. Increased pressures for air pollution abatement make LNG fuel a most promising possibility in the near future for over-the-road vehicles, particularly for public transportation.

In looking into the future, we believe that natural gas or methane, in its frozen solid state or in a "slush" condition, will find unique new markets for natural gas, just as "slush" hydrogen has become important to the aerospace industry. For example, if "slush" or solid methane could be used as a fuel for the SST, the design of the fuel tanks could be simplified to withstand lower G forces during rapid acceleration, and less insulation would be required.

Just as natural gas volume can be reduced by a factor of 630 to 1 when it is liquefied, frozen natural gas can be reduced about 700 to 1. Although solid or "slush" natural gas would be more expensive to produce, it could simplify the transportation of natural gas to remote areas. It would not require attaching or disconnecting cryogenic connecting hoses to move the solid product from a delivery truck to a fuel cell installed in a residential home at a remote location. From a systems viewpoint, this simplified handling could lower the total cost of delivering natural gas for some market applications - including fuel for land, sea, and air transportation equipment.

A systems engineering approach to vehicle propulsion and air pollution control can result in a combination fuel system which uses batteries or fuel cells with a natural gas engine. Surely, many innovations can be expected in this field in the coming decade.

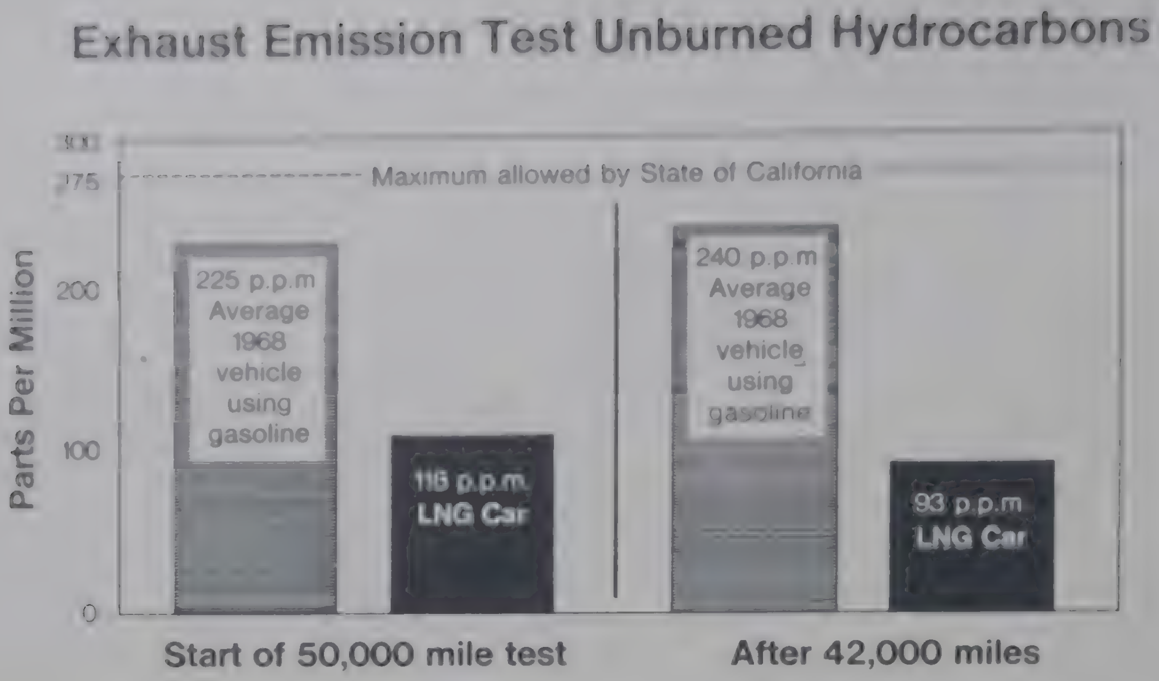
Exhaust Emission Test Unburned Hydrocarbons
Parts Per Million
300
275
200
100
0
Maximum allowed by State of California
225 p.p.m Average 1968 vehicle using gasoline
116 p.p.m. LNG Car
240 p.p.m Average 1968 vehicle using gasoline
93 p.p.m LNG Car
Start of 50,000 mile test
After 42,000 miles

Fig.1

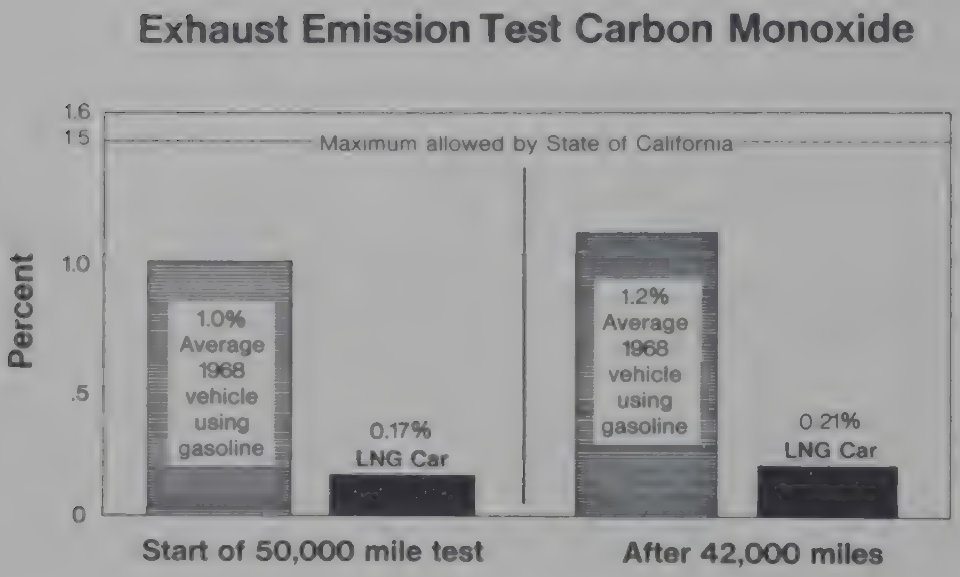
Exhaust Emission Test Carbon Monoxide
Percent
1.6
1.5
1.0
.5
0
Maximum allowed by State of California
1.0% Average 1968 vehicle using gasoline
0.17% LNG Car
1.2% Average 1968 vehicle using gasoline
0.21% LNG Car
Start of 50,000 mile test
After 42,000 miles

Fig.2

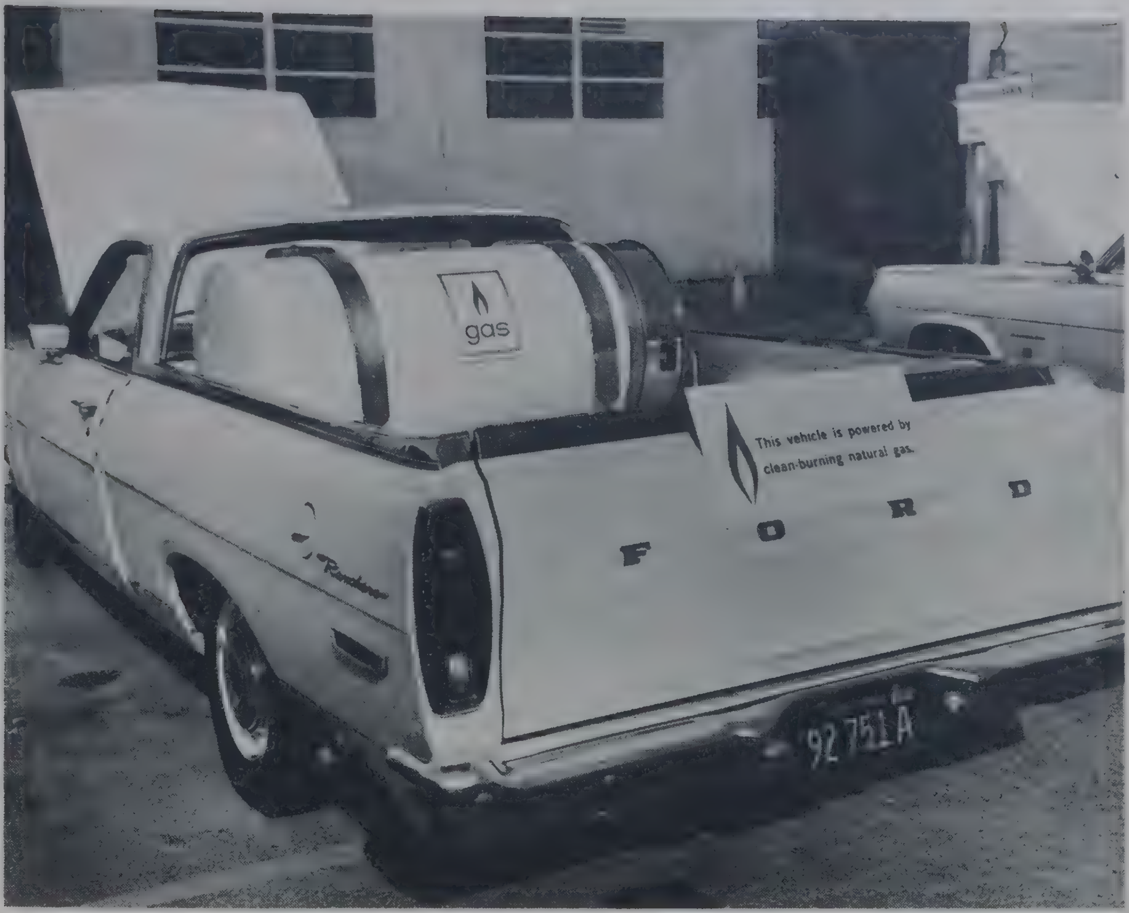
gas
This vehicle is powered by clean-burning natural gas.
FORD
Ranchero
92 751 A

Fig.3

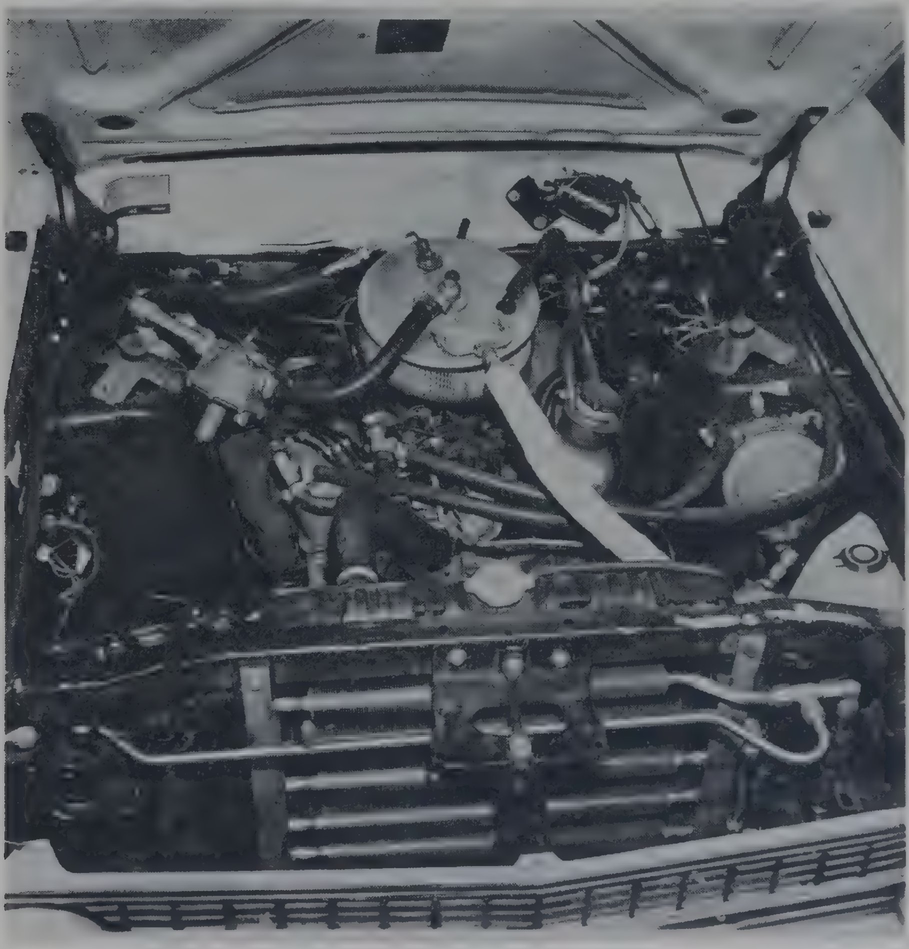

Fig.4

Fig.5

Cryogenic Enterprises Ltd

Fig.6

KENAI PLANT DESIGN (LNG)

By G. E. Thompson and H. R. Sharp

Introduction

Kenai Plant is located about 65 air miles (124 miles by road) southeast of Anchorage, Alaska on the east side of the Cook Inlet and is designed to liquefy natural gas for shipment to Japan. (FIGURE NO. I SHOWS THIS LOCATION). The plant was designed to load a 450,000 barrel tanker every 10 days. The gas supply will be furnished by Phillips Petroleum Company (70%) and Marathon Oil Company (30%), with Phillips operating the plant. A standard cascade liquefaction cycle employing propane, ethylene and methane closed refrigerant systems will be used. Power for driving the refrigerant compressors will be supplied by gas turbines. The on-plot facilities were designed by Phillips Petroleum Company Engineering Department, with the Bechtel Corporation handling the design of off-plot facilities and the construction.

Some Unusual Conditions and Features that Affected the Design

1. The plant supplies utility companies, making continuous operation essential.

A gas distribution company and a large power company will be dependent upon this plant for gas. To supply the maximum reliability and dependability required, only equipment proven by use in LNG service or similar services was used. This contrasts with the normal hydrocarbon processing plants where new innovations frequently are used to reduce investment, realizing that initial use of untried equipment may possibly result in more downtime.

Ships cannot run on an exact schedule and must be loaded when they arrive, to meet delivery commitments. With a 10-day regular shipping schedule this means that very little downtime can be tolerated.

2. Remote Location of Plant.

The plant is located in an area remote from parts stocks and suitable fabrication shops of the specialized equipment used, for emergency work or parts replacement, either during construction or operation. The normal shipping time from Seattle, Washington can be one to two weeks, depending on availability of barges, and another two weeks from factory to Seattle. Thus, equipment had to be purchased earlier than normal to meet job schedules, and more spare parts had to be ordered to be on hand at the site to prevent long downtime on any equipment failure, during startup and operation.

3. Labor and Construction Equipment.

Skilled labor in the Alaska area is in sufficient supply although very costly. It was necessary to establish a camp to provide quarters and subsistence for construction workers not living in the Kenai area.

Warehousing was designed to house and store more than the usual maintenance equipment and spare parts. The rapid industrial development in the Kenai area during the past two years has provided adequate supplies of normal materials and equipment for maintenance and construction. The special equipment parts and material required for an LNG plant still are not available locally and will be warehoused.

4. Severe Weather.

With the lowest recorded temperature of minus 48°F and minimum temperature from minus 20°F to minus 40°F most winters, a design temperature of minus 40°F was selected. This forced the use of special materials for all piping that could be under full pressure at ambient temperatures. These low temperatures also produce a frost line of 8 to 10 feet below grade, which made it necessary to bury fire lines 10 feet deep. Drain cups had to be heat traced to below the frost line, and extensive winterizing of other parts of plant was required, such as pumps, compressors and blowers with gear boxes being located in heated buildings, lube oil stored in heated tanks, and steam tracing of the smaller condensate and water lines subject to freezing.

Winds as high as 80 knots at minus 10°F have been reported. Thus, a design base wind load of 30 psf was chosen for buildings up to 30 feet high, with height and shape factors applied to other structures.

The maximum annual precipitation is only 24.5 inches, but maximum recorded snow depth is 4'-9", so a design snow load of 40 psf was used. This is the snow load recommended by U.S. Army Corps of Engineers for this part of Alaska.

The outdoor construction season is from approximately late March to the middle of November. Receipt of large equipment by barge is limited to the same period due to ice problems in Cook Inlet, and also the Alaskan Highway Department severely limits truck loads during the spring thaws. For these reasons, it was necessary to order the equipment and complete the design much earlier than would otherwise have been required.

5. Type of Material Required.

The low ambient temperatures made it necessary that all plastic and rubber products, which were to be exposed to these temperatures, had to be of suitable materials. This included O-rings, wire insulation, etc. As previously mentioned this also affected the piping materials, requiring the use of materials not normally required. In the process equipment, temperatures down to that of LNG (minus 258°F) were encountered. The special materials required caused some problems, since much of the piping was large, 20" up thru 42". For the cryogenic services stainless steel pipe, valves, and fittings were used along with aluminum exchangers, and aluminum and stainless steel vessels. When the stainless steel flanges mated with aluminum vessel or exchanger flanges, aluminum bolts were used to ensure contraction of the bolts at least as rapidly as the flanges on cool down, to prevent low temperature leaks.

6. Large Size and Complexity of the Plant.

This plant with its large gas turbines (up to 20,000 BHP each) centrifugal compressors, etc., required the use of very large piping. Cooling water headers are 42", compressor suction piping has 42" and 36" lines to handle the larger volumes. The LNG loading line to the dock is 24", and is approximately 2,400 feet long. It is 304 stainless steel, while the compressor piping is all A-333 or A-516 Charpy V-notch tested to minus 50°F and certified. Normal carbon steel pipe was used only in utility service.

The gas turbines were set approximately 15' above grade to accommodate the compressor piping and turbine inlet air ducts under the floor. This allows servicing of units without extensive removal of large piping. With seven gas turbine compressor units the compressor building is approximately 216' x 90'. Figure No. II gives you an idea of the plant size and layout.

Due to the large size, to ensure cleanliness, and to allow more economica piping and insulation, the cold box facilities were built in four units, so that they could be barged to the site.

As an illustration of the design problems caused by the large sizes, consider the need for 30 inch cryogenic quick closing valves to protect the compressors. Phillips and Bechtel independently checked all the available large cryogenic valve manufacturers and then contacted users for their opinions. As information was developed it was exchanged and duplication avoided where possible. Independently we both concluded that for this type service in 20, 24 and 30 inch sizes, and some smaller sizes, the butterfly type cryogenic valves were the best available. Gas operated valve actuators were selected to operate these valves, closing the 30 inch size in less than 10 seconds.

7. Loading Product on Tankers in Harbor with Extreme Tidal Variations and Ice Conditions.

Tidal variations up to 32 feet are experienced. Adding the tidal range to the ship's 10 foot vertical trim range, meant that the loading arms and gangway had to be designed to accommodate a 42' vertical movement, this combined with allowance for a fore and aft run of 15 feet, either side of center, and 10 foot drift away allowance complicated the design. The gangway is pivoted on both the dock and the ship end and has an extension section to allow it to follow the ship, once it is attached. The gangway is positioned onto the ship by an overhead electric powered hoist and trolley. The loading arms, two 12" liquid arms and one 12" vapor return arm, are fully hydraulically powered for positioning, then are free to follow the ship. Limit switches were provided to shut down the loading before the arms reached maximum travel, in case ship should move too far and cause failure of the arms.

The gently sloping beach required that the dock itself be constructed approximately 1,200' from shore to provide adequate water depth. A long span plate girder bridge connected the dock to shore. Strong currents and heavy ice led to the use of a minimum number of large diameter, heavy walled support piling. Each pile was tapered from 10' diameter at mud line to 4' diameter in the tidal zone to minimize the area exposed to floating ice.

8. Location in Earthquake Zone 3.

It was necessary to limit the height of the LNG storage tanks, provide extra tiedowns on tankage, provide extra strength in foundations, restrict the movement of critical piping, etc., in case of tremors or an earthquake. Precautions were taken based on problems encountered in this and other areas due to earthquakes to protect against major damage in all but a catastrophic earthquake for which there is no way to design. Buildings are of the insulated steel sandwich type, instead of masonry. Aluminum was chosen for the LNG storage inner tanks, for its greater ductility at normal operating temperatures of minus 258°F.

Design Responsibilities

These responsibilities were split between the Bechtel Corporation and Phillips Petroleum Company, to allow the plant to be built as quickly as possible and to utilize the special skills and experience of both organizations. Although not covered by this paper, the process design for the entire plant was done by Phillips. Phillips Petroleum Company Engineering Department designed the main plant process facilities at Bartlesville, Oklahoma. Phillips extensive experience in the design and operation of helium extraction plants and purification plants, ethylene plants, refrigerated storage of propane, ethylene, etc., and other low temperature plants was used in the design of these on-plot facilities.

Bechtel Corporation designed the balance of the plant including the inlet gas metering and dehydration, storage and loading, and the utilities. This work was done in their San Francisco, California office, about 1800 miles from Bartlesville. (FIGURE NO. 3 OF THE SIMPLIFIED FLOW SHEET SHOWS THE DIVISION & DESIGN).

Phillips, as Kenai Plant operator, was responsible for coordination of design and procurement of equipment. A Phillips design engineer was assigned to Bechtel's office during the design period. He followed all phases of their design and procurement, calling in other Phillips experts as required. Since Phillips engineerin is done under the Project Team concept, he also served as Phillips Project Team Representative on many matters. Frequent phone calls, exchanges of written informatio and some visits, both ways, were used to keep everyone concerned informed.

All mechanical flow sheets were carefully reviewed by all interested parties, including Phillips process engineers, operating department engineers, design engineers, plant operating personnel and safety engineers, to make sure the plant was being designed to function as desired.

Design Procedures

1. Cold Boxes.

Originally these were laid out as one large cold box. After further study it was decided to split it up into three shop fabricated and assembled cold boxes that could be barged to Alaska, and one small field assembled box containing one large exchanger. Each of the three large boxes contain several exchangers, vessels, valves, etc.

To get the most compact design possible, the boxes were designed by making scale plan and elevation drawings, and concurrently modeling at the 3/8" = 1' scale. As the design proceeded pressure drop and pipe stress calculations were run and changes made accordingly. Both drawings and models were necessary for the extremely congested and complex piping. The models and drawings were used by the fabricator during assembly, and the models are being used to train the plant operating personnel.

The cold boxes were designed in Bartlesville, fabricated and erected 1,500 miles away in a suburb of Los Angeles, California. Designers were sent to the erection site, as needed, to explain, answer questions and make any small changes. They also helped with the inspection and check out of the boxes.

Most of the cold box exchangers are of the brazed aluminim plate fin type. These were built in Indianapolis, Indiana and many were shipped 850 miles to a vessel fabricator where additional fabrication was performed, and then shipped about 1,500 miles to the erector for installation in the cold boxes. After completion, the cold boxes were loaded onto a barge and shipped 4,500 miles to the plant site, unloaded, and then skidded onto their foundations. Temporary bracing, tiedowns, etc., were designed and installed to prevent damage during shipment and handling. These were all removed prior to testing, insulation and placing the plant in operation.

Two areas requiring special study and design were how to seal the cold boxes, including around extended valve stems, and how best to insulate boxes and still be able to get into them in case of trouble. Stainless steel bellows seals were used due to previous problems with elastomer type seals. A combination of Perlite insulation and mineral wool, around critical valves, was the insulation system chosen.

2. Inlet Gas Metering, Dehydration, and Boiler House Areas.

These areas were less congested and complex than the cold boxes, but still complex enough so that they were completely modeled, with only the main equipment located on plan drawings. The models were then sent to the plant site and used by the field to erect the piping. These models will be retained in the plant for training purposes. No piping plans or elevations were drawn, only Pipe Line Erection Diagrams, normally referred to as PLED's. These show one particular line with all dimensions and details required to fabricate the line. It also includes a complete bill of material. PLED's were drawn and used on all lines throughout the plant, including those in the cold boxes.

3. Other Areas.

The piping in the other areas of the plant was designed by making plan drawings and PLED's. This method has the advantage over modeling, only of providing drawings for any future changes or additions, but the disadvantages of being harder for some people to visualize and by not having a model for training purposes.

4. General.

PLED's were made on all piping, partly for keeping account of the material required. Piping with welded connections, fittings, etc. was prefabricated, 4 inch and smaller at the plant site and the larger pipe sizes in pipe fabrication shops. The carbon steel piping was prefabricated at Salt Lake City and shipped about 4,000 miles to the plant site. Most of the stainless steel piping was prefabricated in shops within the Los Angeles area.

Computer Use

1. Line Stresses and Line Sizing.

The computer was used for preliminary line sizing. As soon as the equipment pressure drops were known and layouts were ready, additional line sizing runs were made. These sizes were used with the layouts to calculate line stresses, due to contraction and expansion of the piping and equipment. This resulted in many configuration changes, revised layouts and increased pressure drops (due to added ells). New line size runs were made, sizes increased and further stress calculations run, until pressure drops and stresses were finally both within acceptable limits. These machine runs allowed much more accurate design and saved many weeks of tedious hand calculations.

2. Bulk Materials - Bookkeeping and Ordering.

A preliminary drag of pipe, valves, and fittings was made at the start of the job; this was fed to the computer which wrote the necessary purchase orders. As the design progressed and the PLED's were completed, the material shown on each was fed to the computer. Periodically summaries were run which showed the amount of each item ordered, amount used on each PLED by drawing and line number, and amount remaining or shortage. Whenever shortages became appreciable, a summary was marked up and the computer used to print out additional purchase orders and new summaries. Since the material shown on the PLED's was by Phillips stock number, summaries were run on each PLED. This summary gave a complete description of all items required for the piping on that PLED for the convenience of the warehousemen and pipe fitters.

The computer was used to order miscellaneous electrical material by feeding it the preliminary material lists. As the drawings were completed the amount of material required was manually compiled and this information fed to the computer.

Summaries were run to keep account of material used and as a basis for machine ordering any additional material needed.

The instrument miscellaneous materials were handled similar to the piping, except standard instrument piping detail sheets were used in place of PLED's No individual sheet summaries were run.

3. Simulation Studies.

An analog computer was used to study the compatibility of each gas turbine driver and the power requirements of its compressor, at all the various operating conditions. It was also used to study the instrumentation, bypasses, etc., designed to prevent the compressors from either going into surge, or overloading the gas turbine driver on startup, or automatic shutdown of one of the parallel units. This is especially critical since two large parallel units are used to compress each of the refrigerants and the interruption of any unit reduces the plant capacity by one-half. It is expected that these studies will reduce the problems of compressor operation.

Summation

The Kenai Plant was designed for maximum dependability of operation, ease of operation, and safety. This had to be done in a very limited time to meet construction and startup schedules. Investment had to be kept as low as possible, taking into consideration the various unusual features and conditions. It is expected that the plant will be completed, checked out, and started up in time to have an adequate supply of LNG on hand to load the first tanker when it arrives in July, 1969.

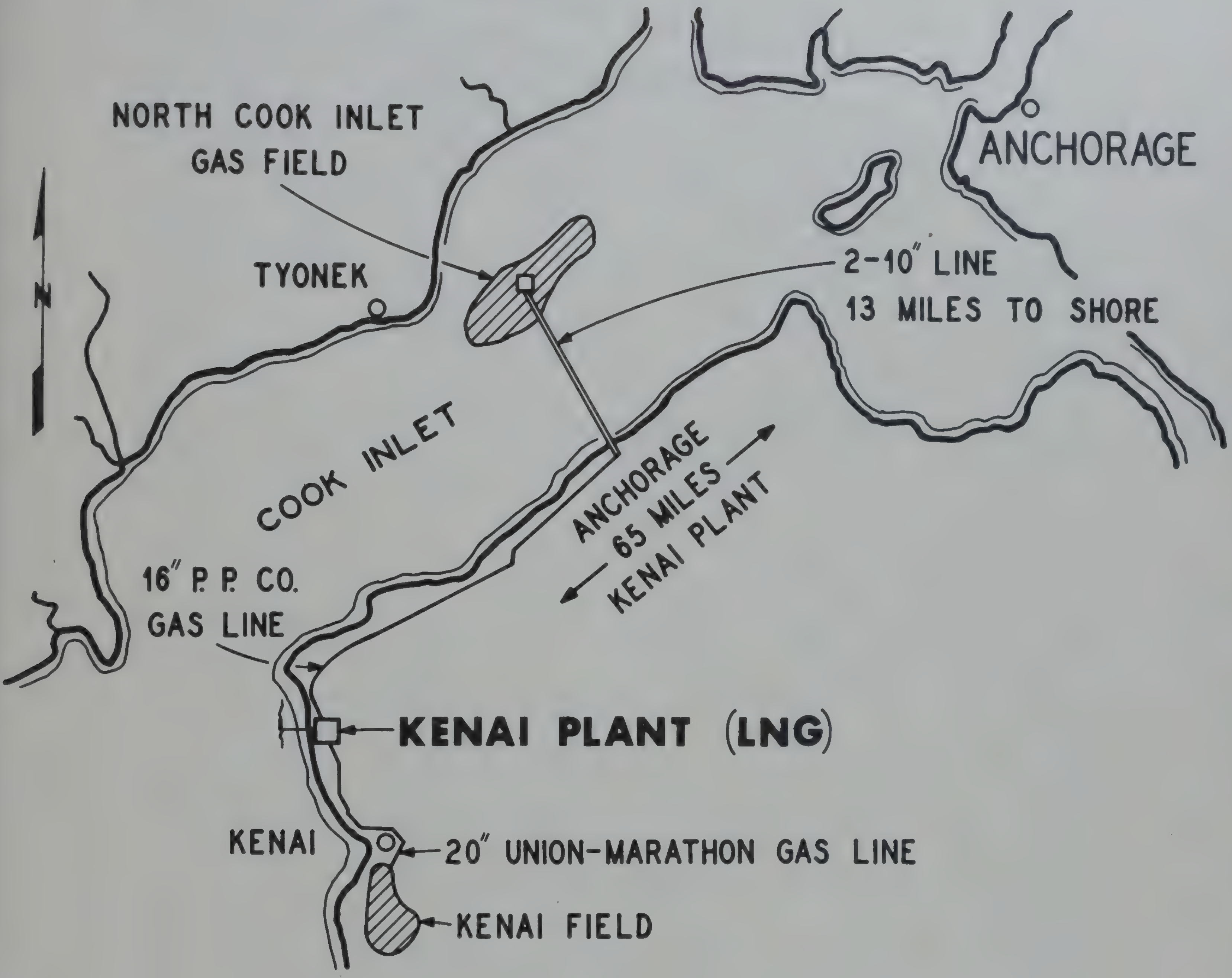
NORTH COOK INLET
GAS FIELD
ANCHORAGE
TYONEK
N
2-10″ LINE
13 MILES TO SHORE
COOK INLET
ANCHORAGE
65 MILES
KENAI PLANT
16″ P. P. CO.
GAS LINE
KENAI PLANT (LNG)
KENAI
20″ UNION-MARATHON GAS LINE
KENAI FIELD

Fig.1. Location of Kenai plant (LNG)

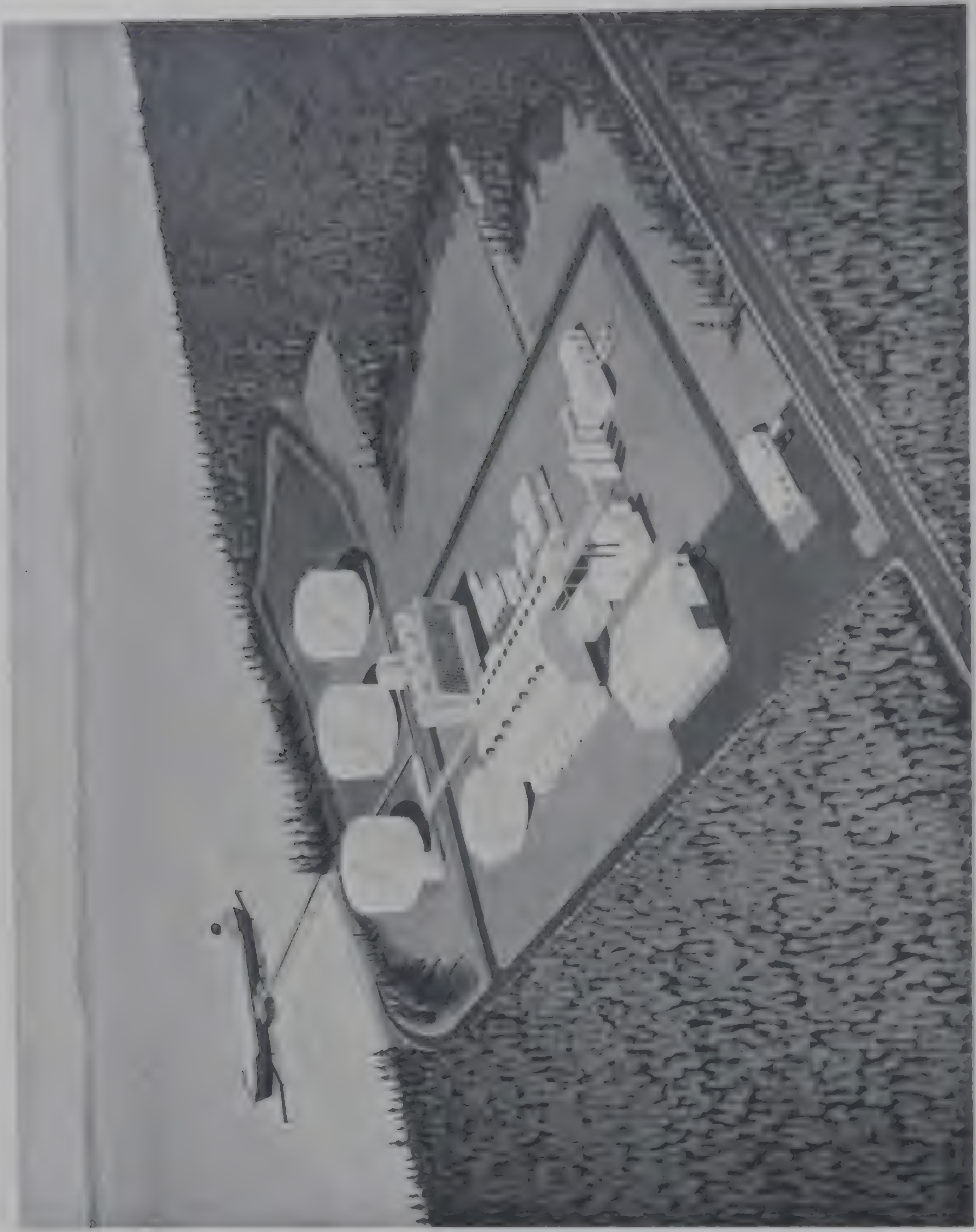

Fig.2. An artist's conception of Kenai plant

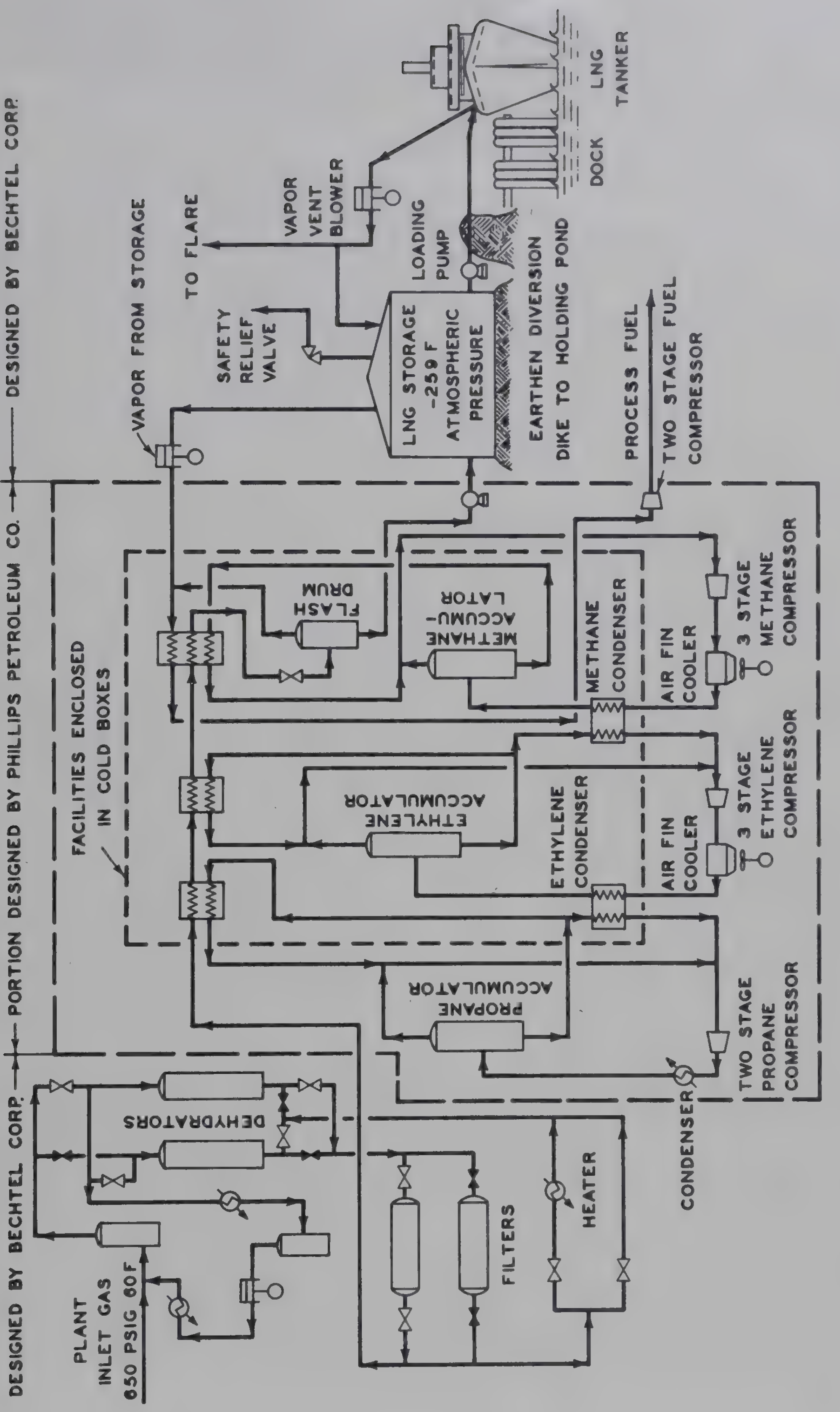
DESIGNED BY BECHTEL CORP.
PORTION DESIGNED BY PHILLIPS PETROLEUM CO.
DESIGNED BY BECHTEL CORP.
PLANT INLET GAS 650 PSIG 60F
DEHYDRATORS
FILTERS
HEATER
CONDENSER
TWO STAGE PROPANE COMPRESSOR
PROPANE ACCUMULATOR
ETHYLENE CONDENSER
AIR FIN COOLER
3 STAGE ETHYLENE COMPRESSOR
ETHYLENE ACCUMULATOR
FACILITIES ENCLOSED IN COLD BOXES
METHANE CONDENSER
AIR FIN COOLER
3 STAGE METHANE COMPRESSOR
METHANE ACCUMULATOR
FLASH DRUM
TWO STAGE FUEL COMPRESSOR
PROCESS FUEL
VAPOR FROM STORAGE
LNG STORAGE -259 F ATMOSPHERIC PRESSURE
SAFETY RELIEF VALVE
TO FLARE
VAPOR VENT BLOWER
LOADING PUMP
EARTHEN DIVERSION DIKE TO HOLDING POND
DOCK
LNG TANKER

Fig.3. Simplified flow - Kenai plant (LNG)

Calculation and Experimental Determination of Enthalpies of Natural Gas Mixtures

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There is an increasing demand for larger and more perfectly developed plant facilities for purification, liquefaction and storage of natura gas. Unfortunately there are only very few reliable experimental data or calculation methods known for the thermodynamic and physical properties of natural gases at cryogenic temperature.

Inaccurate data necessitate the addition of safety factors in the design of the liquefaction cycle and equipment - causing higher cost that could be saved if more accurate data would be available.

This will be demonstrated by an example.

The exergy loss in a heat exchanger depends on the finite temperature differences of the heat exchanging systems.

(Fig. 1)

In the past natural gas liquefaction plants worked with cascades of pure gases as refrigerants which often caused big temperature differences in the heat exchangers meaning great exergy losses.

Modern liquefaction plants work with a single refrigerant consisting of a gas mixture with a similiar cooling curve as the natural gas, thus reducing the temperature differences and consequently the exergy losses in the heat exchangers. The design of heat exchangers with small temperature differences needs very accurate cooling curves, respectively enthalpy values of the natural gas and the gas mixtures. Hence experimental research is required for a start to provide the necessary data which are the base for reliable correlations and predictive techniques.

At first three methods of computing the enthalpy of mixtures are discussed as typical examples of the numerous more or less applicable methods. Then a high pressure isobaric flow calorimeter is described for the experimental determination of the enthalpy values.

The Calculation of the Enthalpy of a Gas mixture

A very simple approximation method is demonstrated in Fig. 2. For example the enthalpy value of a nitrogen-methane mixture has to be determined.

1. The enthalpy of a N_2-CH_4 mixture at 200 °K and 200 ata

   The enthalpy of the pure N_2 and pure CH_4 will be read at 200 ata and 200 °K, multiplied by the mole fraction and added.

   The enthalpy of the pure components will be read at the partial pressure, multiplied by the mole fraction (or not) and added.

   Both methods are more or less inaccurate. The exact enthalpy value of the mixture lies in between.

2. The enthalpy of the N_2-CH_4 mixture at 140 °K and 20 ata in the vapor phase

   Under these conditions pure N_2 is in the supercritical but pure CH_4 in the liquid state. The enthalpy of the pure N_2 can be read. Pure CH_4 does not exist in the vapor phase under these conditions. We can help ourselves by extrapolating the isobar from the vapor phase region into the two phase region of pure methane. Moreover, it is uncertain if it should be done at the total pressure or at the partial pressure of methane.

3. The enthalpy of the N_2-CH_4 mixture at 140 °K and 20 ata in the liquid phase

   The enthalpy of the pure CH_4 can be read from the diagram. But pure N_2 does not exist in the liquid phase under these conditions. Again the enthalpy of the N_2 in the liquid phase of the mixture can be read on the rectilinear extension of the boiling curve into the supercritical state of the pure N_2.

Although there is some thermodynamic justification for this method no high accuracy can be expected.

Other approximation methods work with charts of generalized thermodynamic properties of gases and liquids. These properties are plotted as functions of reduced pressures $p_r = p/p_c$ for different values of reduced temperatures $T_r = T/T_c$.

The equation of corresponding states is prepared by averaging data of a group of gases which represents only a special molecule type or an average of many molecule types

$$p_r = \left(\frac{RT_c}{p_c V_c}\right)\left(\frac{T_r}{V_r}\right) \quad F\,(V_r, \; T_r) \tag{1}$$

Figure 3 shows the reduced enthalpy deviation of an actual gas from the ideal gas versus the reduced pressure for different reduced temperatures by the Hougen-Watson method. This method is a good approximation for a mixture taken as a single component with critical constants equal to the molar averages

$$T'_c = \sum_i x_i \, (T_c)_i \tag{2a}$$

$$p'_c = \sum_i x_i \, (p_c)_i \tag{2b}$$

Here x_i is the mole fraction of the i-th component.

However, this treatment of mixtures gives only good results in the vapor phase.

Numerous empirical equations of state have been proposed for describing the p-V-T behavior of gases and liquids. In general these equations are only suitable for a particular application. One of the best proved equations of state is still the well-known Benedict-Webb-Rubin equation.

$$p = RT\,\frac{1}{V} + (B_oRT - A_o - C_o/T^2)\,\frac{1}{V^2} + (bRT-a)\,\frac{1}{V^3} + a\alpha\,\frac{1}{V^6} + \left(c\,\frac{1}{V^3\,T^2}\right)\left(1 + \gamma\,\frac{1}{V^2}\right)\exp\left(-\gamma\,\frac{1}{V^2}\right) \qquad (3)$$

It can be applied to both vapor and liquid phases of hydrocarbons when its eight parameters have been selected accordingly. It can be extended to cryogenic conditions by varying the C_o- and γ-coefficients with temperature.

The enthalpy of liquid pure light hydrocarbons can be calculated from the equation down to a temperature of 100 °K (= -280 °F) within an accuracy of ± 2,5 J/g (≈ ±1 Btu/lb). For hydrocarbon mixtures and systems containing nonhydrocarbons the BWR equation can also be used for the calculation of enthalpies as well as other properties of mixture.

In the attempt to use a generalized equation of state certain compromises must be made between the simplicity of the mathematical calcule and the accuracy of the results.

A generalized equation of state gives the possibility to compute the thermodynamic data, e.g. density, enthalpy, entropy, equilibria of any single or multi-component system based on one single equation, possibly one system of mixing rules and one set of as few as possible constants for each component.

As on the one hand the interactions of molecules in mixtures differ for various mixtures and as on the other hand only one set of constants should be used, a certain inaccuracy is unavoidable.

The highest degree of accuracy, that means the optimum set of constants, can be obtained if a maximum number of experimental values is available.

In order to utilize these experimental data a computation program is needed which allows direct corrections of the constants based on experimental values. However, there are still two areas where all present techniques for enthalpy calculations of mixtures are grossly in error:

1. the low temperature compressed liquid region and
2. the region in the vicinity of the critical point.

In spite of the highly developed equations of state the improvement of the constants used in the equations and the failure of the equations in the certain critical areas require intensive experimental work.

For designing LNG plants MGG uses both the computer calculated data based on the BWR equation and the experimentally determined thermodynamic data.

Experimental Determination of Enthalpy

MGG obtains experimental enthalpy values in the gas phase by using a Joule-Thomson device. A high pressure isobaric flow calorimeter was built in order to extend the experimental facilities to include the two phase region as well as the liquid phase.

Principle of Operation

In Figure 4 a schematic flow diagram of the calorimeter is shown. In a two stage diaphragm compressor the testgas is compressed up to 170 atmospheres with a maximum throughput of 7 Nm^3/h and is stored in a high pressure vessel. Then it flows through a heat exchanger where the testgas will be precooled by evaporating nitrogen and the returning gas. In a following evaporator cryostat the testgas is brought to the inlet temperature of the calorimeter by controlled evaporation of nitrogen. The fluid enters the calorimeter. Here an exactly controlled and measured amount of heat is added electrically. After temperature and pressure measurement the fluid is brought to room temperatur by a controlled water bath and throttled down to 1,5 atmosphere befor entering the flowmeters. Then it passes to the low pressure vessel flows back to the compressor for recirculation.

Apparatus

The calorimeter with the main equipments is shown in detail in Fig. 5. The calorimeter consists of a porous bronze sinter compound which has to be passed by the fluid. The sinter compound allows good heat exchanging and mixing of vapor and liquid in the two phase region. The sinter compound is shrinked up in a cylindrical copper block which is surrounded by the heating coil.

The evaporator cryostat is designed similiar to the calorimeter. The heating coil is replaced here by a tube wound round the copper block. The evaporating nitrogen is pumped through the tube by means of a vacuum unit.

The calorimeter, the evaporator and a heat exchanger are housed in an evacuated container (Fig. 6). The vacuum is maintained at 10^{-5} Torr by continuous pumping.

Flow Measurement and Control

The gas is throttled down to a lower pressure for the flow measurement Then is passes a wet gas meter. A special low vapor-pressure oil instead of water has been used to fill the meter, in order to avoid evaporation of water into the testgas. The error in measurement by the wet gas meter is about ± 1 %.

For controlling the continuous and undisturbed flow further flow meter are installed. A rotameter and an orifice can be calibrated by the wet gas meter.

The pressure of the testgas in the calorimeter is controlled by a pneumatically-loaded dome pressure controller. The low pressure for the flow measurement is adjusted by spring-loaded pressure controllers. The pressure is read on Bourdon tube precision gauges with an accuracy of ± 0.1 %.

Different sizes of pressure gauges are employed to keep the maximum error in measurement for all pressure values lower than + 0.5 %.

The inlet temperature of the testgas stream for the calorimeter is held constant by heat exchanging of the testgas and evaporating nitrogen in the evaporator cryostat. An electronic temperature controller settles the nitrogen flow through the cryostat by a magnetic valve. The inlet and outlet temperatures of the calorimeter are measured by platinum resistance thermometers. The temperatures are read on a five decades potentiometer free of thermo-voltage. The thermometers have been calibrated by aid of a calibration platinum resistance thermometer in a specially designed cryostat.

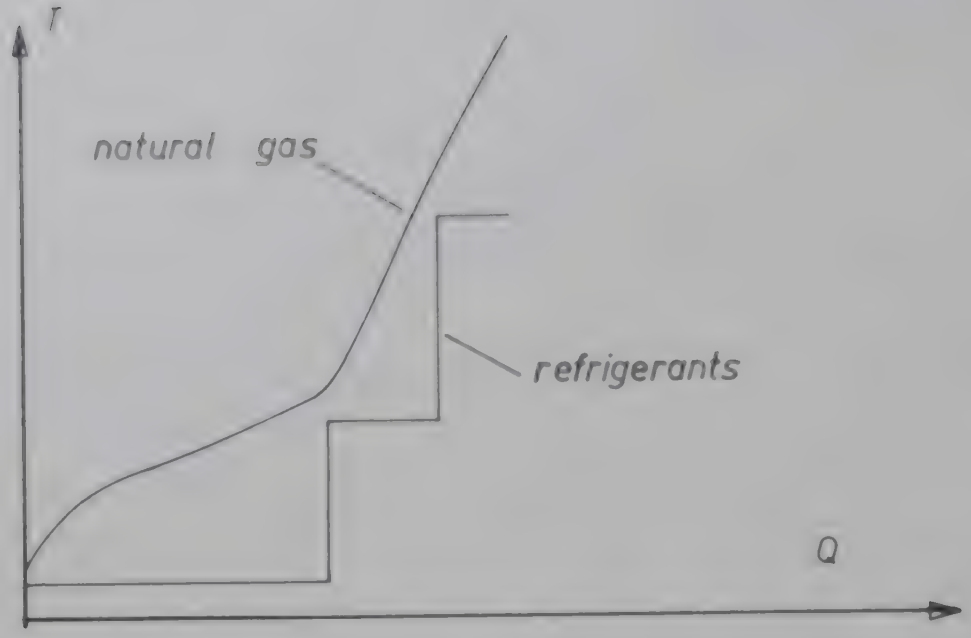
The calibration thermometer was calibrated by Physikalisch-Technische Bundesanstalt (PTB). The accuracy of the temperature measurements is better than ± 0.05 centigrade.

Experimental Results

The last picture shows the 40 atmosphere isobar of a mixture of 6 % N_2, 84 % CH_4, 7 % C_2H_6 and 3 % C_3H_8, this is the composition of the main components of the Algeric natural gas.
The isobar is computed by the BWR equation and some points are measured by the Joule-Thomson device in the gas phase with good agreement.

At this moment we are testing and calibrating the calorimeter and then we will have the first measurement runs for obtaining experimental enthalpy values in the two phase and liquid region, too.

The calorimeter and the Joule-Thomson device will enable MGG to determine experimentally all interesting enthalpy values for the design of LNG plants and to apply the BWR equation of state with more accuracy.


T
natural gas
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Q

Fig. 1a: cooling curves for natural gas liquefaction by pure gases as refrigerants

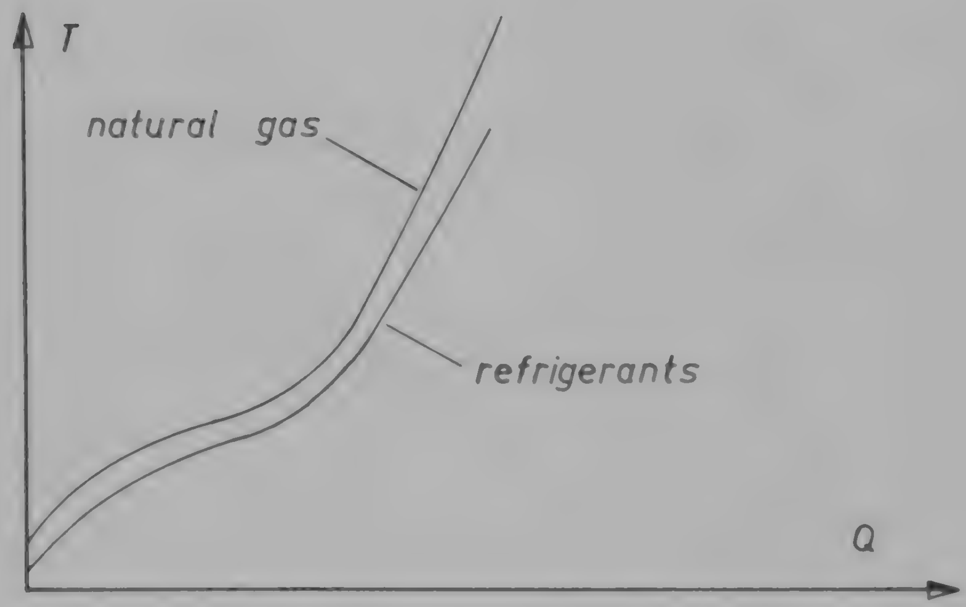
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Fig. 1b: cooling curves for natural gas liquefaction by gas mixture cascades as refrigerants

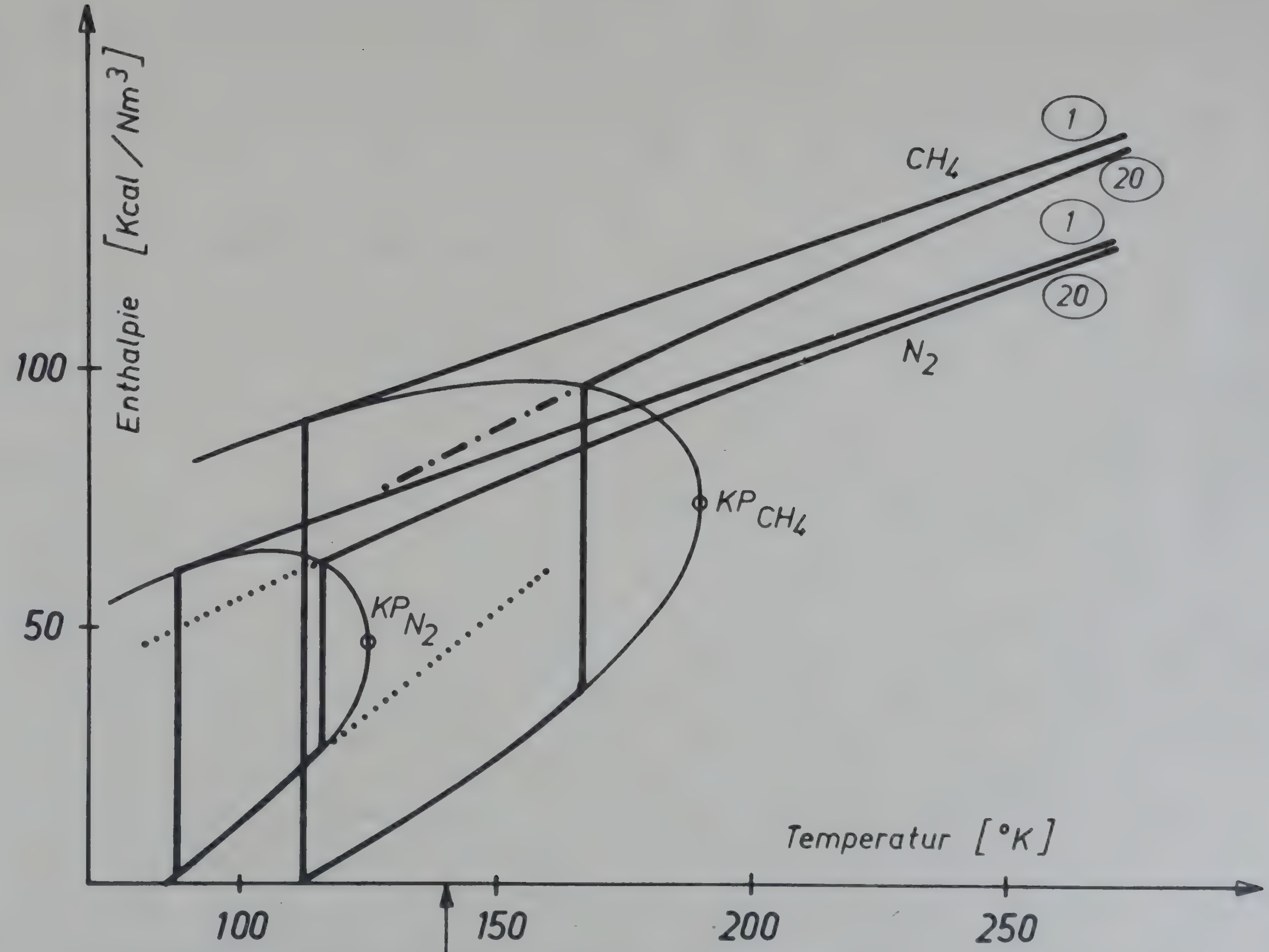


Fig.2: Enthalpy-temperature diagram for N_2 and CH_4

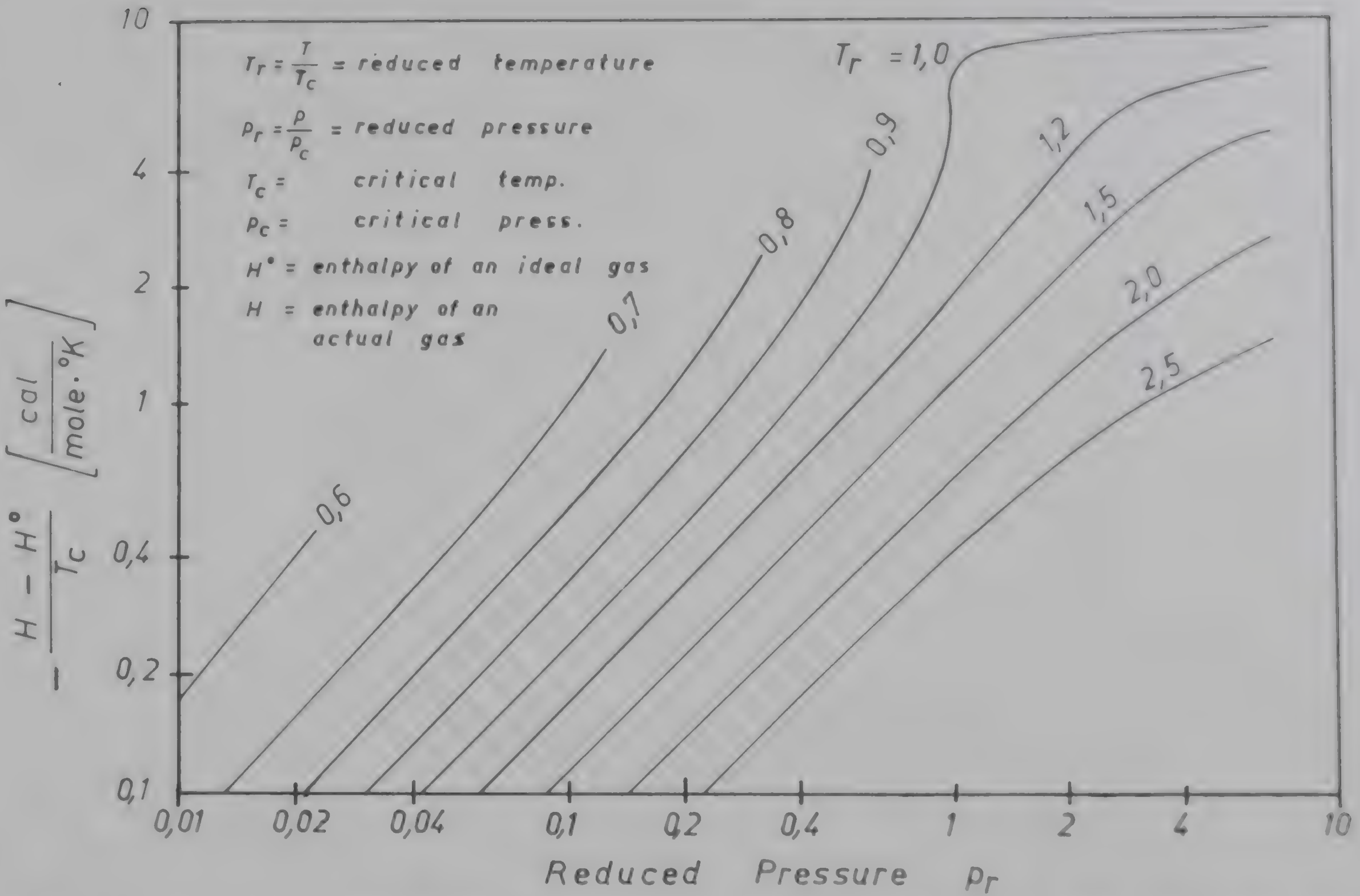


Fig. 3: The enthalpy of gas imperfection as a function of the reduced pressure and the reduced temperature

[From O. A. Hougen and K. M. Watson, Chemical Process Principles, Wiley (1947), Part II]

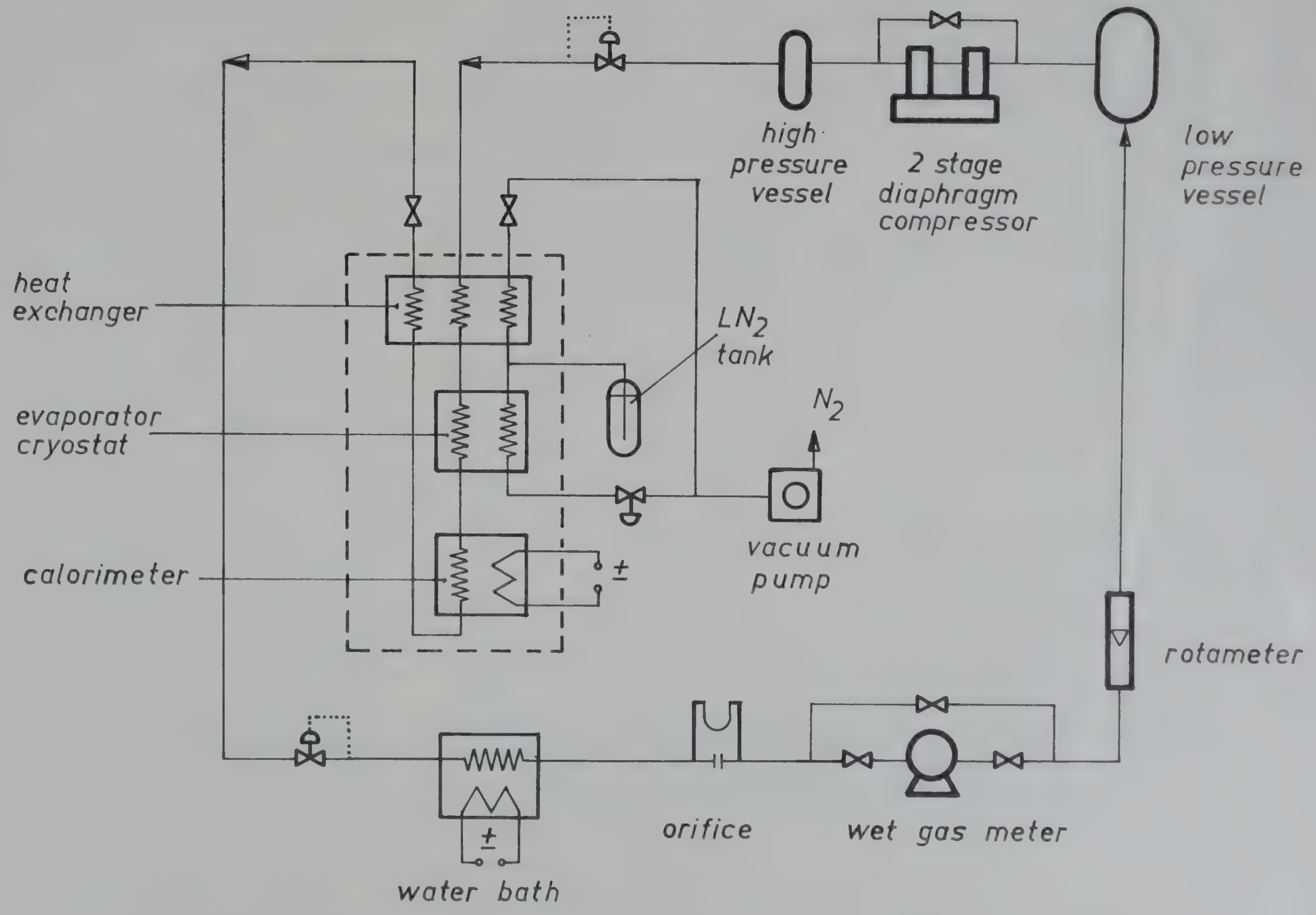
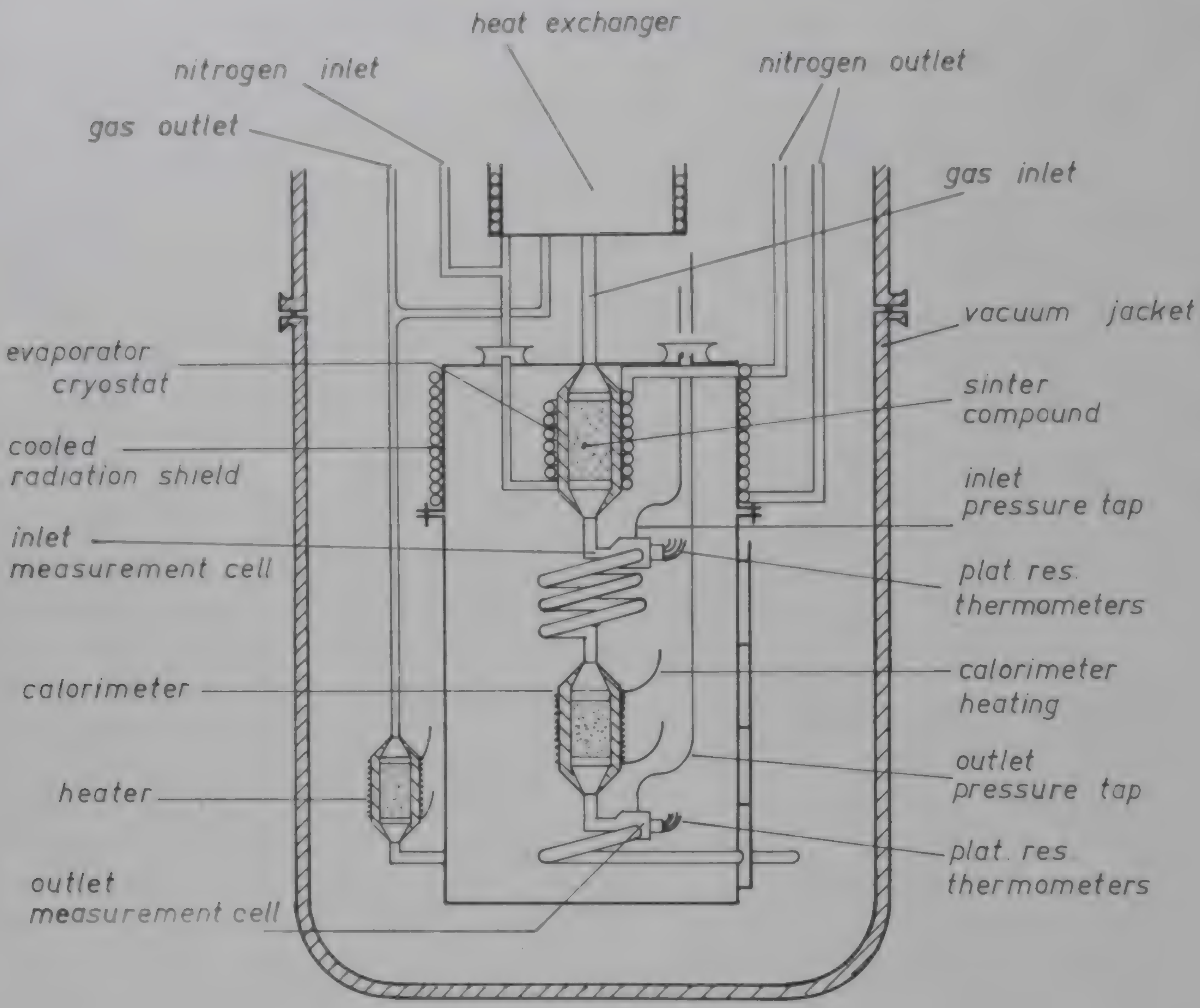

high pressure vessel
2 stage diaphragm compressor
low pressure vessel
heat exchanger
LN_2 tank
N_2
evaporator cryostat
vacuum pump
calorimeter
±
rotameter
orifice
wet gas meter
±
water bath

Fig. 4: Schematic flow diagram of the calorimeter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nitrogen inlet
nitrogen outlet
gas outlet
gas inlet
vacuum jacket
evaporator cryostat
sinter compound
cooled radiation shield
inlet pressure tap
inlet measurement cell
plat. res. thermometers
calorimeter
calorimeter heating
heater
outlet pressure tap
plat. res. thermometers
outlet measurement cell

Fig. 5: Cross – section of the Calorimeter

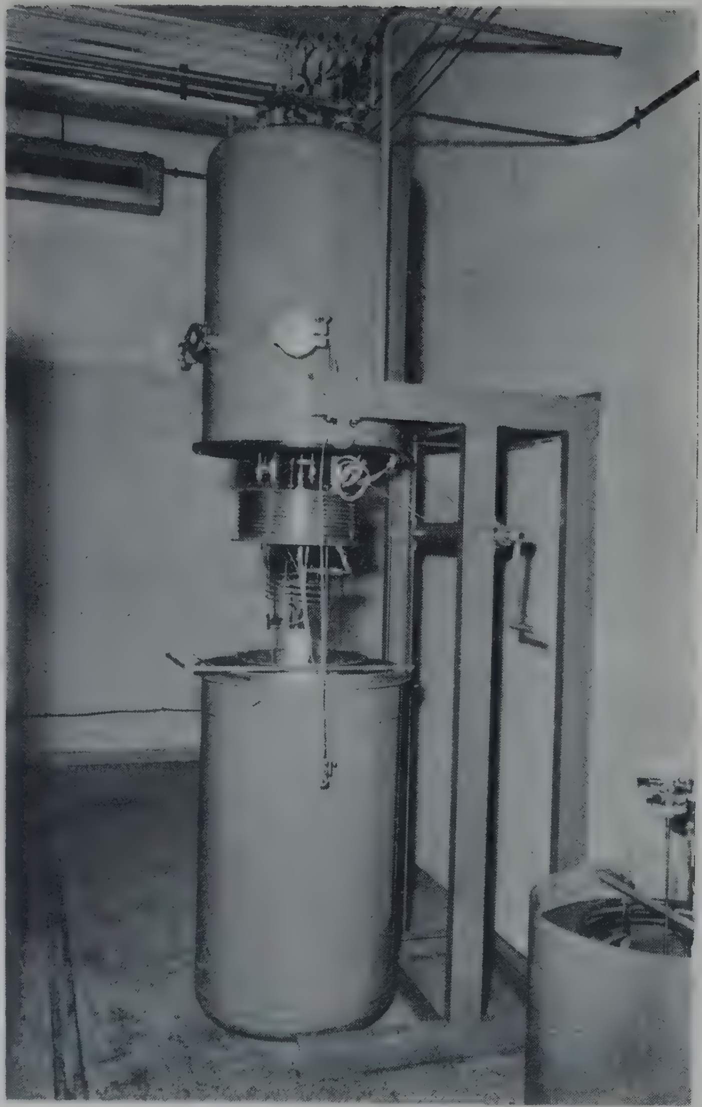

Fig.6

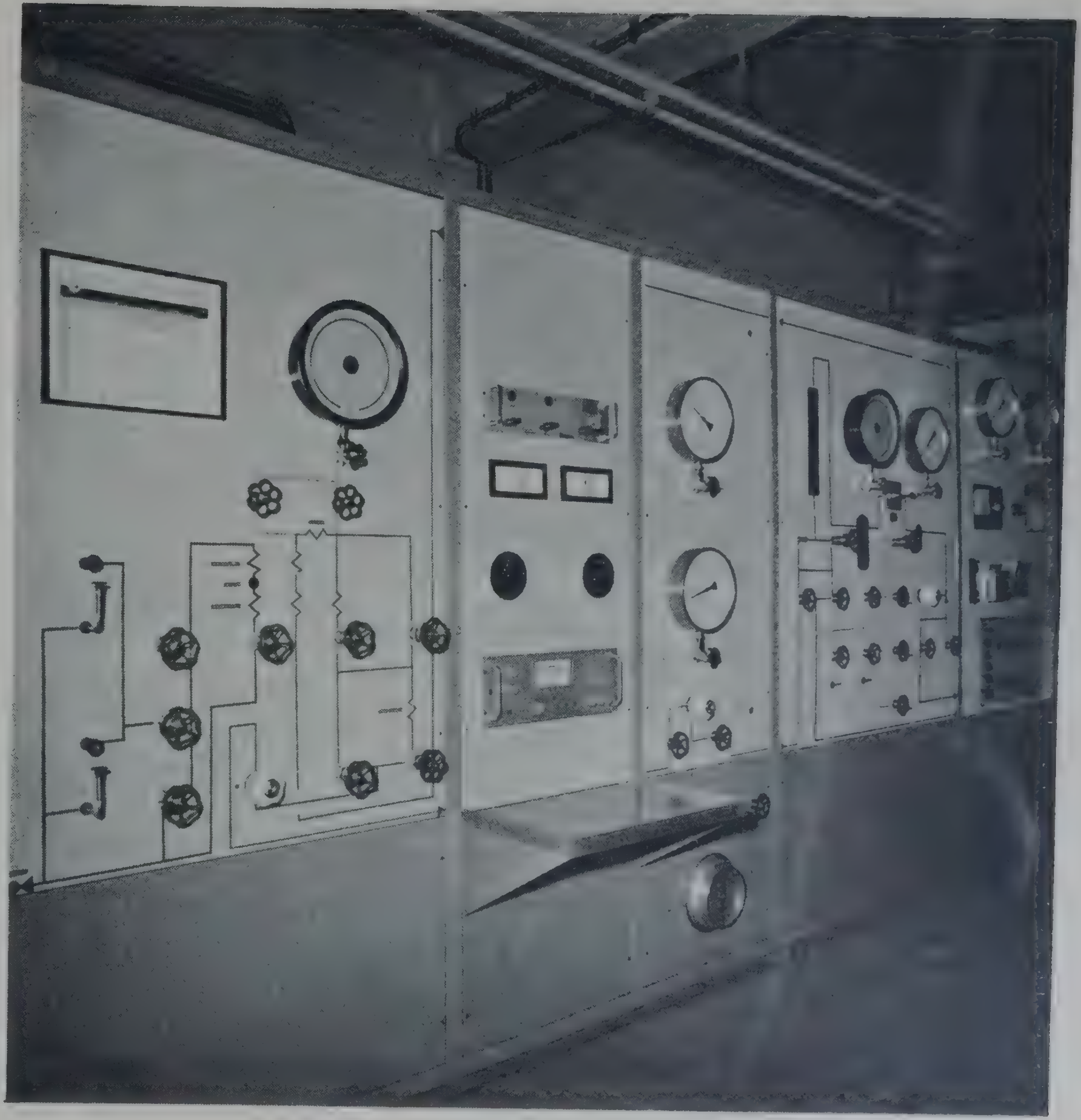

Fig.7

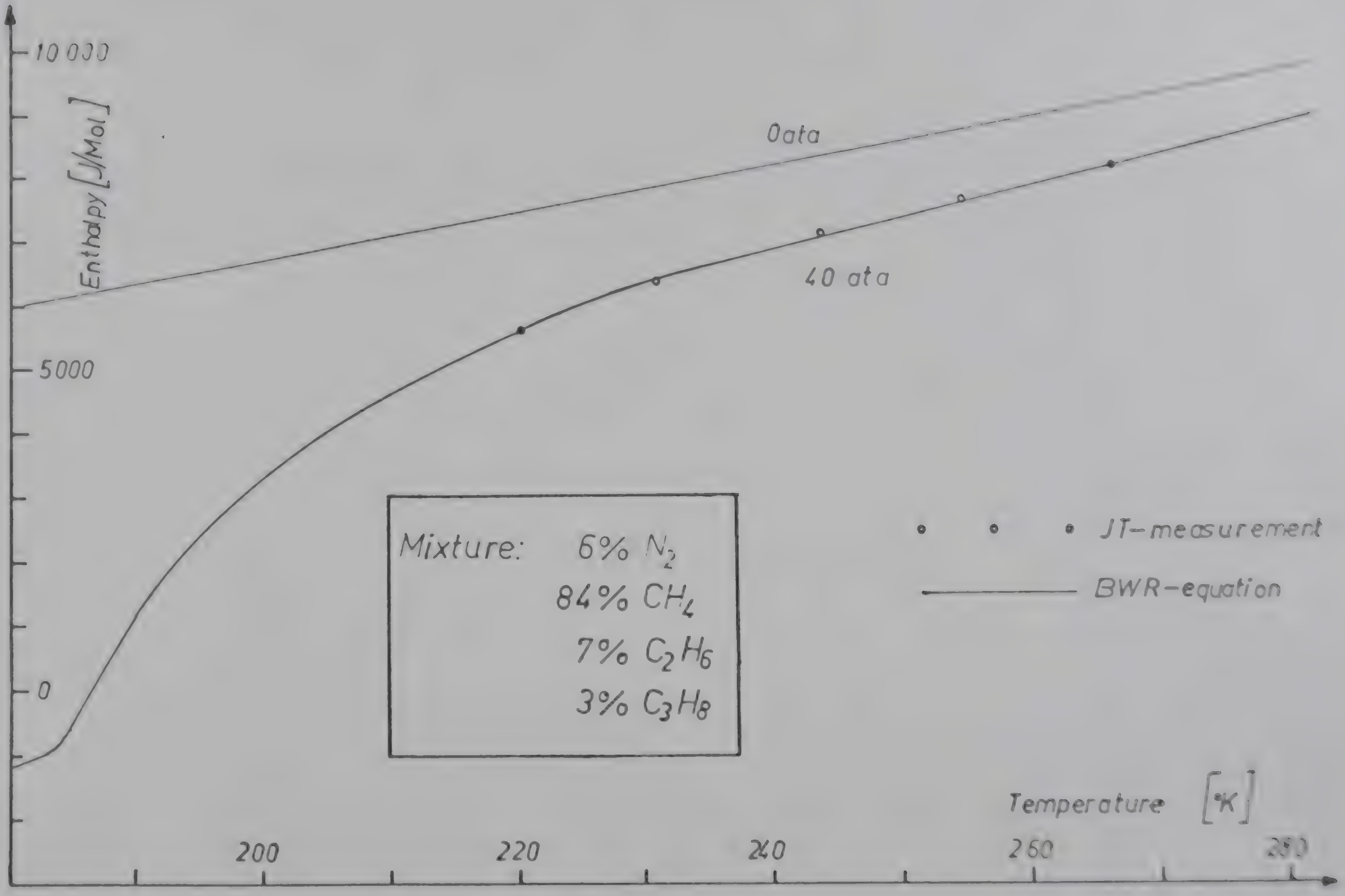

Fig. 8: Enthalpy-temperature diagram

The Measurement and Prediction of Enthalpies and Phase Equilibria in Natural Gas Mixtures

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(Department of Chemical Engineering
University of Leeds)

Synopsis

A description is given of a flow calorimeter suitable for pressures up to 100 atmospheres in which the feed gas is cooled from a starting temperature of 100°C to final temperatures in the range +10 to -165°C. Heat is absorbed by liquid oxygen through a variable thermal link. Provision is made for sampling and analysing vapour and liquid samples at outlet from the calorimeter. The accuracy of the enthalpy measurements is ± 0.7%.

Results are reported for a single simulated North Sea Gas mixture containing slightly more than 90% methane for 2 isobars at 28 and 38 atm. approximately, and temperatures down to -120°C. When these results are compared with predictions based on the Redlich Kwong, Hildebrand, and Grayson and Streed equations it is found that good correspondence occurs at temperatures down to those at which about 5 % liquid is condensed.

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Introduction

Previous studies by Tilley[1] and Nathan[2] in the Authors' laboratory had resulted in the development of a calorimeter with an accuracy of ± 2% and capable of separating samples of the two phases when they were present at the outlet pressure and temperature. The chromatograph used by Nathan was capable of analysing these samples to an accuracy of only ± 3% of total composition. During this time Sahgal et al[3] described a simple variable thermal link for use in this type of calorimeter, and it was decided therefore to redesign the apparatus using this device but retaining some of the proven components of Nathan's design, especially the type of cooling coil and sample separator.

The new calorimeter was designed for a maximum working pressure of 100 atmos. and a temperature range spanning from a constant inlet temperature of approximately 100°C to an outlet temperature of between 20°C and -165°C. Contrary to the practice adopted by most other workers, the coolant used was liquid oxygen. There were three reasons for this choice. During a temporary shut-down a test gas rich in methane would not freeze, the impurities in the coolant would not tend to accumulate in the calorimeter (if liquid nitrogen was used, any oxygen impurity would accumulate leading to a temperature and latent heat variation), and finally the latent heat per unit volume is 51% higher than for nitrogen. The last point was important because it was desired to be able to run the apparatus continuously for up to 8 hours using a single 5 gallon liquid oxygen Dewar. The throughput of the apparatus (approximately 12 scfh) was related to this criterion, and it was also limited by the output of the circulating gas compressor (20 scfh). The oxygen hazard was taken into account in the apparatus design.

Calorimeter Design

In addition to the general design specifications mentioned in the introduction the following criteria were taken into account.

1. The total heat inleak (along the inlet and exit tubes, and from the environment) was to be kept to within 2% of the total heat transferred, and a correction applied for it based on a calibration with pure nitrogen gas over the operating temperature range.

2. The total heat capacity of the cold zone of the calorimeter was to be kept to a minimum so that the apparatus would reach a steady condition quickly, and so that the required length of run (which must be long enough to render negligible thermal effects due to residual temperature drifts) would not be more than, say, 1 hour.

3. A turbulence promoter would be used in the calorimeter tube (after Tilley[1]) so that good contact is achieved between liquid and vapour, leading to steady condensation and the production of reliable samples.

The detailed design of the calorimeter is shown in fig. 1.

The feed gas, already preheated, is brought to a constant temperature of approximately 100°C by an electric heater contained within the entry tube. Control is applied by means of a platinum resistance thermometer placed just below this heater. The inlet temperature is finally measured by a copper-constantan thermocouple contained in a stainless-steel hypodermic tube, and mounted with the exposed junction level with the elbow connection to the calorimeter coil.

The coil contains 3 unsupported turns which serve as a heat break to minimise conduction losses to the main calorimeter coil. This coil (containing 100 inches of 10 s.w.g. stainless-steel hypodermic tubing) is soft-soldered to a thin walled copper tube which conducts heat to the base of the variable thermal link. It also contains throughout its length, a single 28 s.w.g. stainless-steel wire, loosely knotted at intervals of $\frac{1}{2}$ inch, to act as a turbulence promoter.

From the calorimeter coil the test fluid enters the sample trap which has a capacity of about 2 ml. Most of it leaves through a central tube which extends about two thirds of the way down the trap and terminates with a chamfer. Thus, with a 2-phase system, liquid tends to collect in the lower part of the trap until its surface is broken by the outlet tube, then both phases flow out together. In this way small reservoirs of separated gas and liquid are created from which samples can be withdrawn. The calorimeter outlet temperature is measured by a thermocouple which is inserted down the gas sampling tube into the trap until its junction is opposite the side entry tube. The disentrainment of droplets from gas samples is assisted by the presence of the thermocouple in the sampling tube. A thermal bridge is achieved between all the exit tubes immediately above the sample trap by wrapping with copper wire for a length of 1 inch.

The variable thermal link is made of stainless steel, and its shape provides for a control ratio of about 10 : 1 . It is moved by a retractable jack which is disengaged during normal running.

The liquid oxygen feed serves to cool the outlet vapour pipes before it cascades, via the cooling trough on top of the liquid reservoir, into the guard vessel. The feed is controlled by a solenoid valve actuated by a heated thermistor bead sensing device. A constriction was placed in the blow-off line from the guard vessel, and was adjusted so that the liquid levels in the guard and reservoir were kept at all times to within $\frac{1}{5}$ inch of each other (as indicated by an external manometer).

The calorimeter coil was surrounded by a copper radiation shield which conducted its heat to the liquid oxygen reservoir; and this shield was in turn surrounded by a second one which absorbed all radiation from the outer vessel and conducted it to the guard vessel. A few layers of super-insulation (layers of aluminium foil separated by thin layers of glass cloth or tissue paper) were installed on the coil, radiation shields and various tubes, as shown in the diagram. The outer vessel and the inner surfaces of the shields were also lined with aluminium foil. In this way cold losses were minimised and the loads placed on the two radiation shields were kept small.

The flow circuit is shown in fig. 2. The feed gas was contained in a storage tank of 9 ft^3 capacity at the intermediate pressure (approximately 100 psig) 2-stage Corblin Diaphragm Compressor. By this means the volume of storage was kept low without the risk of condensation. From the upper stage the test gas mixture was passed through driers (KOH pellets, silica gel, and finally molecular sieve type 4A) to a needle valve A controlling the flow. The supply pressure was controlled by a precision regulating valve K and other needle valves G and J which by-passed the gas back to the intermediate pressure.

The pressure of the test gas in the calorimeter was controlled by the needle valve B and the precision pressure regulator C, both submerged in an oil bath maintained at room temperature. The flow was measured in a positive displacement type gas meter operating at very near atmospheric pressure (determined by the gas holder). Both the gas meter and gas holder were filled with low vapour pressure silicone oil.

Analysis

The chromatograph was required to analyse paraffin hydrocarbons from methane through to pentane together with nitrogen and hydrogen. For the last two components argon was used as the carrier gas in a 3 ft. column packed with activated charcoal. For the hydrocarbons helium was used in a 35 ft. column packed with Squalene on graded firebrick. Katharometer detectors were used in both streams, and the outputs were brought together onto a Kent Chromalog integrator.

The chromatograph was calibrated using binary mixtures of all components with methane, the composition of the test mixtures being obtained from direct specific gravity determinations. The estimated final accuracy of the composition values was ± 0.6% of the actual value or - 0.05% of total composition, whichever is the larger.

Determination of heat leakage

Nitrogen gas (oxygen-free grade, B.O.C. Ltd.) was passed through the apparatus at 2 different flowrates and at 2 different pressure levels. The enthalpy data of Din[6] was used for this gas, the accuracy being about ± 0.7%. The latent heat of oxygen at atmospheric pressure was taken as 1629 cal/mole. The heat leak (taken as the difference between the heat gained by the oxygen and that liberated by the nitrogen) is plotted against outlet temperature in fig. 3. In general this heat inleak represented about 2% of the total heat being transferred, and so came within the design criterion.

The results were correlated by regression analysis giving the following equation :-

$$Q_L = 201.84 - 2.8889\ t$$

where

Q_L is the heat leak in cal/hr

t is the exit test gas temperature in °C

The root mean square deviation was 45 cal/hr (from 0.1 to 0.3% of the total heat transferred). The heat leak was not found to change significantly with the level of vacuum though this varied between 6.5×10^{-4} and 1.6×10^{-3} mm Hg.

Operating Procedure

During apparatus commissioning considerable difficulty was experienced due to surging of the coolant oxygen gas flow each time the solenoid valve in the liquid supply line opened. At first this was attributed to gas leakage from the outer guard vessel into the inner oxygen reservoir. However it was then appreciated that a change in the relative liquid levels in the guard and inner vessels would cause gas displacement. Precautions were therefore taken to minimise the surges by improving the insulation of the feed line and by careful control.

In general after starting from warm it took $1\frac{1}{2}$ hours to bring the apparatus to steady conditions. An experiment was generally run at steady conditions for a further $1\frac{1}{2}$ hours during which regular readings of temperatures, pressures and flows were taken. The gas meters, together with a clock, were photographed simultaneously at appropriate intervals. Normally each enthalpy measurement was completed before the two phases were sampled alternately through the chromatograph. Both samples flowed continuously during this period, small cuts being taken from them for analysis.

Results

Enthalpy and phase equilibrium measurements were carried out on a simulated natural gas with a composition representative of North Sea gas, containing Nitrogen, Methane, Ethane, Propane and iso-Butane. These measurements covered the range -20° to -120°C on two isobars at 28 and 38 atmospheres. A total of 54 measurements were made, 20 of which were at the lower pressure, concentrating primarily on the two phase region.

The mixture was made up in the storage tank which had been evacuated after previously containing nitrogen from the calibration experiments. The hydrocarbon gases used were Philips Petroleum pure grade (99%), any impurities being predominently the light paraffins present in this mixture.

The choice of iso-Butane in preference to n-Butane was dictated primarily by the time taken to analyse mixtures. Using the 35 ft. 'Squalene' column, n-Butane requires an extra 5 minutes to reach the detector compared with the isomer, increasing the time for an analysis to 20 minutes.

The overall mixture was analysed whenever possible during and between experimental runs and this revealed a gradual change in composition.

The amount of nitrogen apparently increased due to adsorption of hydrocarbons by the large silica gel drier inserted in the flow circuit. The initial and final mixture composition is given below.

	Initial run %	Final run %
Nitrogen	3.65	4.25
Methane	90.66	90.65
Ethane	4.08	3.69
Propane	1.24	1.06
iso-Butane	0.37	0.35

The variation in composition was quite gradual and gives confidence in the accuracy of the analyser.

The results of all experiments are presented in Table 1.

Phase equilibria measurements are given in Table 2 for only a few runs because difficulty was experienced in obtaining consistent consecutive analyses of the same phase. For some runs the phases were analysed over a period of 3 hours, and although the resulting enthalpy measurements were consistent over 45 minute intervals, and the average outlet temperature remained steady within 0.1°C, the phase results were very unsteady. The greatest difficulty lay with the liquid analysis which varied as much as ± 12% of the peak areas. Although the vapour phase results were more reproducible (about ± 6% of the chromatogram peak areas), they were not as consistent as single phase analyses in the vapour region (± 0.5% of the chromatogram peak areas).

Another problem lay in the determination of the liquid boundary because very small amount of non-condensibles (H_2 and Helium ?) were present, and accumulated in the vapour sample stream. The fact that the sample flows were less than 0.5% of the total flow accentuated this difficulty.

Experimental Accuracy

Estimates of accuracy for the various experimental readings are as follows :-

Pressure ± 2 psi (± 0.1 atm)

Temperature at inlet ± 0.12°K, at outlet ± 0.13°K

Flow measurement (the gas meters were calibrated against each other and against a soap-film meter) ± 0.15% for the test gas, ± 0.12% for Oxygen.

The influence of these and other inaccuracies on the determination of the enthalpy change for run 16 are given in the following table

Measurement	Error produced in cal/mole
Inlet temperature	± 1.2
Outlet temperature	± 3.4
Pressure	± 3.3
Flow of test gas	± 3.6
Flow of oxygen	± 2.9
Oxygen latent heat	± 0.5
Heat leak correction	± 3.6
Total maximum error	±18.5 cal/mole

Since the total heat transferred was approximately 2500 cal/mole this represented an error of ± 0.7% . In fact repeat experiments in this range were found to be reproducable to ± 0.25%.

Prediction of Phase Equilibria

Effort was concentrated on the procedure previously described by the Authors[4], based on Shelton and Wood[5], which uses the Redlich Kwong equation for the vapour fugacities, the Hildebrand solubility parameter to give liquid activity coefficients, and the Grayson and Streed equation for liquid fugacity coefficients. Values of the liquid fugacity coefficient for Nitrogen were provided separately by Shelton. Other constants used are given in Table 3.

Whilst the computer program converged quite rapidly and gave results of tolerable accuracy just inside the dew point region (see Run 37, Table 2) rapid deterioration set in as the amount of the liquid phase increased. In the case of the 4 runs at around 28 atm. pressure Runs 8 and 15, at which the experimental results indicated about 25% of the feed had liquefied, the predicted amount of condensation was only 14%. A drop of temperature of about 2°K to Run 16 showed increased condensation to 37% whilst the computer program showed no significant change. With a further drop of temperature of about 3°K to Run 9 condensation had increased to 74% and the program would no longer converge.

Amongst the 5 runs at pressures around 38.4 atm. a similar bahaviour is apparent. There is a temperature span of only 17.9°K between runs 39 and 29 whilst the condensation changes from 4 to 87%. At this pressure the program could be made to converge on a stable solution even for this low experimental vapour fraction but the calculated vapour fraction is wildly in error.

It is apparent that a gas mixture of this type, having a very high concentration of one of the most volatile components, places extreme demands on the prediction method. Near to the dew point condensation proceeds gradually over a temperature range of about 10°K. But thereafter condensation occurs very rapidly, almost isothermally, until the mixture is nearly all liquid. The location of this narrow temperature zone is clearly beyond the scope of the prediction method. The difficulty of obtaining reproducible phase samples in this region was also explained since very small pressure or temperature variations would lead to gross changes in the composition of the phases. The enthalpy measurements, depending on integrated flow readings over time intervals of more than 30 minutes, were not so sensitive.

It is probable that each part of the prediction method was being used beyond its valid range. Difficulties were experienced in obtaining a real root to the Redlich Kwong equations especially at stages in the convergence procedure when the assumed values of V and the equilibrium ratios yielded a fictitious vapour composition far removed from the actual solution. The Grayson and Streed correlations used to give the liquid phase fugacity coefficients were also being applied at temperatures below those for which they were derived.

Prediction of Enthalpy

For the vapour phase the isothermal enthalpy correction H^V can be derived from the Redlich Kwong equation, as

$$\Delta H^V = RT\left(-1.5\frac{A^2}{B}\ln(1+h) + Z - 1\right)$$

Hence the total enthalpy of the vapour H^V_{mix} will be given by

$$H^V_{mix} = \sum y_i\ H^T_{oi} + \Delta H^V$$

The standard state enthalpies H^T_{oi} of the components have been presented as 6th order polynomials by Shelton[7] from the tabulated values of Rossini.

The isothermal enthalpy correction for the liquid phase may be expressed in terms of the Grayson and Streed fugacity coefficients and the acentric factors for each component by the following equation :

$$-\frac{\Delta H_i^L}{RT^2} = \left(\frac{\partial \ln \nu_i^{(o)}}{\partial T}\right)_P + w_i\left(\frac{\partial \ln \nu_i^{(1)}}{\partial T}\right)_P + \left(\frac{\partial \ln \gamma_i}{\partial T}\right)_{P,\, x_i}$$

where
$$\frac{1}{2.303}\left(\frac{\partial \ln {}_i^{(o)}}{\partial T}\right)_P = -\frac{C1}{T_r^2} + C2 + Z\ C3T_r + 3\ C4T_r^2 + \left(C6 + 2C7T_r\right)P_r + C9\ P_r^2$$

and
$$\frac{1}{2.303}\left(\frac{\partial \ln \nu_i^{(1)}}{\partial T}\right)_P = -9.45672\ T_r^2 + \frac{1.22060}{T_r^2} + 8.65808$$

$$\left(\frac{\partial \ln \gamma_i}{\partial T}\right)_{P,x_i} = -1.8\left(\frac{V_i^L(\bar{\delta} - \delta_L)^2}{RT^2}\right)$$

then $H^L_{mix} = \sum x_i \ H^T_{oi} + \sum x_i \Delta H^L_i$

The predicted enthalpy change was found by calculating values of H_{mix} at both the inlet and outlet temperatures for each calorimeter run at the prevailing pressure. The difference between the predicted and measured enthalpy changes, expressed as a percentage deviation, is given in the final column in Table 1.

It is seen that enthalpy changes involving the vapour phase alone are all within ± 2% and mostly within 1%. This accuracy is also sustained for liquid fractions up to about 7%, but further into the condensation region the deviation may exceed 20%. These errors are no doubt largely due to the faulty prediction of V . For the all-liquid region the predicted enthalpy change is always less than the measured one and the deviation varied from 4.7 to 13.4%.

Conclusions

Results with an accuracy of ± 0.7% are presented for the enthalpy changes experienced by a simulated North Sea Gas mixture cooling from + 100°C to — 120°C at pressures of approximately 28 and 38 atmospheres. Some phase equilibrium measurements are also given. It is shown that with a gas of this type, containing more than 90% CH_4, condensation, particularly in the range 20 to 95% liquid, occurs over a very narrow temperature range.

Using a prediction method based on the Redlich Kwong, Hildebrand and Grayson and Streed equations acceptable phase data are provided only for liquid fractions up to about 5%. Enthalpy predictions based on the same approach are generally within ± 1% in the vapour region and for similar small quantities of liquid.

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Acknowledgements

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Symbols

A, B	constants in the Redlich Kwong equation.
C1, C2, etc.	Constants in the Grayson and Streed correlation.
h =	Modified Redlich Kwong Constant BP/Z.
H^V_{mix} and H^L_{mix}	Enthalpies per mole of mixture in the vapour and liquid phases.
H^T_{oi}	standard state enthalpy of component i at temperature T.
P_r and T_r	reduced pressure and temperature.
y_i and x_i	mole fractions of component i in the vapour and liquid phases.
$\bar{\delta}$, δ_L	Hildebrand solubility parameters.
γ_i	activity coefficient of component i.
ν_i	liquid phase fugacity coefficients.

TABLE 1. Enthalpy measurements for a simulated natural gas mixture

RUN No.	FEED GAS COMPOSITION N2	C1	C2	C3	i-C4	PRESSURE atm.	AVERAGE TEMPERATURE °K IN	OUT	ENTHALPY CHANGE Cal/mole	PREDICTED VAPOUR FRACTION	% DEVIATION IN H
1	3.65	90.66	4.08	1.24	0.37	28.0	373.36	255.14	1147.4	VAP	+0.32
2	3.66	90.67	4.07	1.23	0.37	28.0	373.22	238.57	1311.1	VAP	+0.01
3	3.67	90.66	4.07	1.23	0.37	28.1	373.75	222.09	1468.5	VAP	+1.06
4	3.69	90.64	4.07	1.23	0.37	28.1	373.87	204.32	1732.7	0.971	+0.22
5	3.76	90.58	4.07	1.22	0.37	28.0	373.92	212.68	1614.2	0.989	-0.36
6	3.78	90.56	4.07	1.22	0.37	28.0	373.91	219.95	1515.2	VAP	-0.59
7	3.80	90.56	4.06	1.21	0.37	28.0	373.08	192.95	1941.8	0.935	-1.36
8	3.81	90.56	4.05	1.21	0.37	28.1	373.27	181.33	2324.7	0.863	-7.19
9	3.83	90.57	4.03	1.20	0.37	28.1	373.11	174.92	2982.9	0.789	-21.96
10	3.86	90.61	3.99	1.17	0.37	28.0	373.05	196.80	1841.1	0.952	-0.02
11	3.87	90.64	3.96	1.16	0.37	28.2	372.05	196.75	1843.4	0.952	-0.07
12	3.89	90.66	3.94	1.14	0.37	28.1	372.97	196.83	1843.9	0.953	-0.39
13	3.90	90.68	3.91	1.14	0.37	28.0	372.99	196.69	1842.0	0.953	-0.25
14	3.91	90.70	3.89	1.13	0.37	28.3	373.38	193.19	1916.1	0.940	-0.61
15	3.93	90.72	3.86	1.11	0.37	27.6	372.62	179.37	2367.4	0.861	-8.56
16	3.94	90.74	3.84	1.11	0.37	27.4	372.73	177.61	2495.8	0.846	-11.71
17	3.95	90.76	3.83	1.10	0.36	27.4	373.03	202.43	1737.4	0.973	-0.35
18	3.95	90.80	3.80	1.10	0.35	28.2	372.40	156.50	3563.0	LIQD	-10.00
19	3.95	90.81	3.79	1.09	0.35	28.2	372.47	170.93	3293.7		
20	3.96	90.81	3.79	1.09	0.35	28.4	372.43	190.12	1990.4	0.930	-2.34
21	3.97	90.80	3.79	1.09	0.35	38.5	372.28	196.38	2032.2	0.926	-2.36
22	3.97	90.83	3.78	1.08	0.34	38.4	372.11	193.16	2161.4	0.909	-5.03
23	3.99	90.84	3.76	1.07	0.34	38.4	372.08	192.71	2185.2	0.907	-5.68
24	3.99	90.84	3.76	1.07	0.34	38.8	372.38	191.78	2238.3	0.899	-6.32
25	4.00	90.83	3.76	1.07	0.34	38.9	372.40	191.75	2247.9	0.899	-6.55
26	4.01	90.82	3.76	1.07	0.34	38.9	372.28	191.81	2232.2	0.899	-6.03
27	4.01	90.83	3.75	1.07	0.34	38.7	372.29	191.97	2206.6	0.901	-5.35
28	4.03	90.82	3.75	1.06	0.34	38.3	372.59	219.31	1578.3	0.998	-0.55
29	4.04	90.81	3.75	1.06	0.34	38.3	372.83	184.25	2871.8	0.841	-18.49
30	4.04	90.81	3.75	1.06	0.34	38.3	372.80	184.30	2855.6	0.841	-18.14

cont'd

TABLE 1. Enthalpy measurements for a simulated natural gas mixture

RUN No.	FEED GAS COMPOSITION N2	C1	C2	C3	i-C4	PRESSURE atm.	AVERAGE TEMPERATURE °K IN	OUT	ENTHALPY CHANGE Cal/mole	PREDICTED VAPOUR FRACTION	% DEVIATION IN H
31	4.04	90.81	3.75	1.06	0.34	38.3	373.51	227.27	1500.1	VAP	-1.17
32	4.05	90.80	3.75	1.06	0.34	38.4	373.68	233.96	1423.6	VAP	-0.98
33	4.06	90.79	3.75	1.06	0.34	38.5	373.10	187.88	2467.1	0.871	-10.60
34	4.07	90.78	3.75	1.06	0.34	38.4	373.07	187.98	2452.2	0.872	-10.28
35	4.08	90.77	3.75	1.06	0.34	38.2	372.99	146.20	3687.3	LIQD	-13.40
36	4.10	90.76	3.74	1.06	0.34	38.3	373.26	173.44	3238.2	LIQD	-5.01
37	4.11	90.75	3.74	1.06	0.34	38.2	372.34	173.56	3228.1	LIQD	-4.68
38	4.13	90.74	3.73	1.06	0.34	38.5	373.26	232.47	1434.2	VAP	-0.84
39	4.14	90.73	3.73	1.06	0.34	38.2	373.36	201.86	1891.0	0.954	-1.08
40	4.15	90.73	3.72	1.06	0.34	38.2	373.30	201.93	1882.7	0.955	-0.82
41	4.16	90.73	3.71	1.06	0.34	38.3	373.32	188.93	2371.7	0.882	-8.60
42	4.17	90.72	3.71	1.06	0.34	38.7	373.33	185.44	2716.0	0.851	-15.36
43	4.18	90.73	3.69	1.06	0.34	38.4	373.24	235.40	1399.3	VAP	-0.78
44	4.19	90.72	3.69	1.06	0.34	38.5	373.17	186.70	2579.6	0.863	-13.13
45	4.19	90.72	3.69	1.06	0.34	38.6	373.23	186.61	2605.1	0.862	-13.75
46	4.20	90.70	3.69	1.06	0.35	38.6	373.20	186.63	2600.2	0.862	-13.58
47	4.21	90.69	3.69	1.06	0.35	38.5	373.19	186.75	2579.7	0.863	-13.17
48	4.22	90.68	3.69	1.06	0.35	38.3	373.25	183.29	2936.6		
49	4.22	90.68	3.69	1.06	0.35	38.6	373.33	179.74	3145.5		
50	4.23	90.67	3.69	1.06	0.35	38.6	373.31	179.93	3121.7		
51	4.24	90.66	3.69	1.06	0.35	38.6	373.25	179.93	3118.3		
52	4.25	90.65	3.69	1.06	0.35	38.50	373.27	180.16	3108.8		
53	4.25	90.65	3.69	1.06	0.35	38.5	373.26	245.76	1299.2	VAP	-1.62
54	4.25	90.65	3.69	1.06	0.35	38.4	372.66	254.07	1201.0	VAP	-1.33

TABLE 2. Experimental and Predicted Phase Equilibria Values for Selected Runs.

Run	TEMP	PRESSURE		COMPOSITION OR K VALUE					VAPOUR
No.	°K	atm.	SYMBOL	N2	C1	C2	C3	i-C4	FRACTION
8	181.33	28.1	y	4.65	93.84	1.40	0.10	0.01	0.756
			x	1.05	80.21	12.45	4.79	1.50	
			K (exp)	4.39	1.17	0.112	0.0213	0.0072	
			K (pred)	8.29	1.37	0.078	0.0124	0.0044	0.863
9	174.92	28.1	y	8.30	91.06	0.59	0.04	0.01	0.261
			x	2.08	90.13	5.51	1.73	0.55	
			K (exp)	4.02	1.00	0.107	0.0281	0.0226	
15	179.37	27.6	y	4.92	93.52	1.31	0.21	0.04	0.731
			x	1.18	82.81	10.86	3.88	1.27	
			K (exp)	4.14	1.12	0.126	0.0538	0.0299	
			K (pred)	7.81	1.34	0.073	0.0114	0.0040	0.861
16	177.61	27.4	y	5.38	93.49	1.00	0.11	0.02	0.632
			x	1.34	85.69	8.94	3.04	0.99	
			K (exp)	4.03	1.09	0.112	0.0335	0.0169	
			K (pred)	7.20	1.30	0.069	0.0107	0.0038	0.846
23	192.71	38.4	y	4.57	92.94	2.18	0.27	0.04	0.836
			x	1.31	79.80	11.82	5.18	1.89	
			K (exp)	3.49	1.16	0.184	0.0518	0.0440	
			K (pred)	5.05	1.36	0.125	0.0260	0.0111	0.907
29	184.25	38.3	y	7.73	91.20	0.94	0.11	0.02	0.128
			x	3.10	91.02	4.34	1.21	0.33	
			K (exp)	2.48	0.997	0.217	0.0875	0.0517	
			K (pred)	2.93	1.226	0.122	0.0293	0.0144	0.841
33	187.88	38.5	y	5.45	92.94	1.44	0.15	0.02	0.606
			x	1.87	87.31	7.39	2.56	0.87	
			K (exp)	2.91	1.06	0.195	0.0557	0.0253	
			K (pred)	3.69	1.28	0.117	0.0255	0.0114	0.872
39	201.86	38.3	y	4.30	91.96	3.10	0.54	0.10	0.958
			x	0.77	61.59	17.71	13.38	6.55	
			K (exp)	4.75	1.49	0.174	0.0405	0.0141	
			K (pred)	10.75	1.54	0.153	0.0312	0.0125	0.955
41	188.93	38.3	y	5.23	92.95	1.61	0.18	0.03	0.692
			x	1.72	85.46	8.55	3.15	1.12	
			K (exp)	3.03	1.09	0.188	0.0546	0.0249	
			K (pred)	3.98	1.297	0.118	0.0251	0.0110	0.882

TABLE 3

Value of Constants used in Redlich Kwong-Hildebrand-Grayson and Streed equations

COMP	Tc °R	Pc psia	w_i	$_i$ $(cal/ml)^{\frac{1}{2}}$	v_i^L ml/mole
N_2	227.2	492.3	0	2.58	36
C1	343.9	673.1	0	5.68	52
C2	550.0	709.8	0.1064	6.05	68
C3	665.9	617.4	0.1538	6.40	84
i-C4	734.7	529.1	0.1825	6.73	105.5

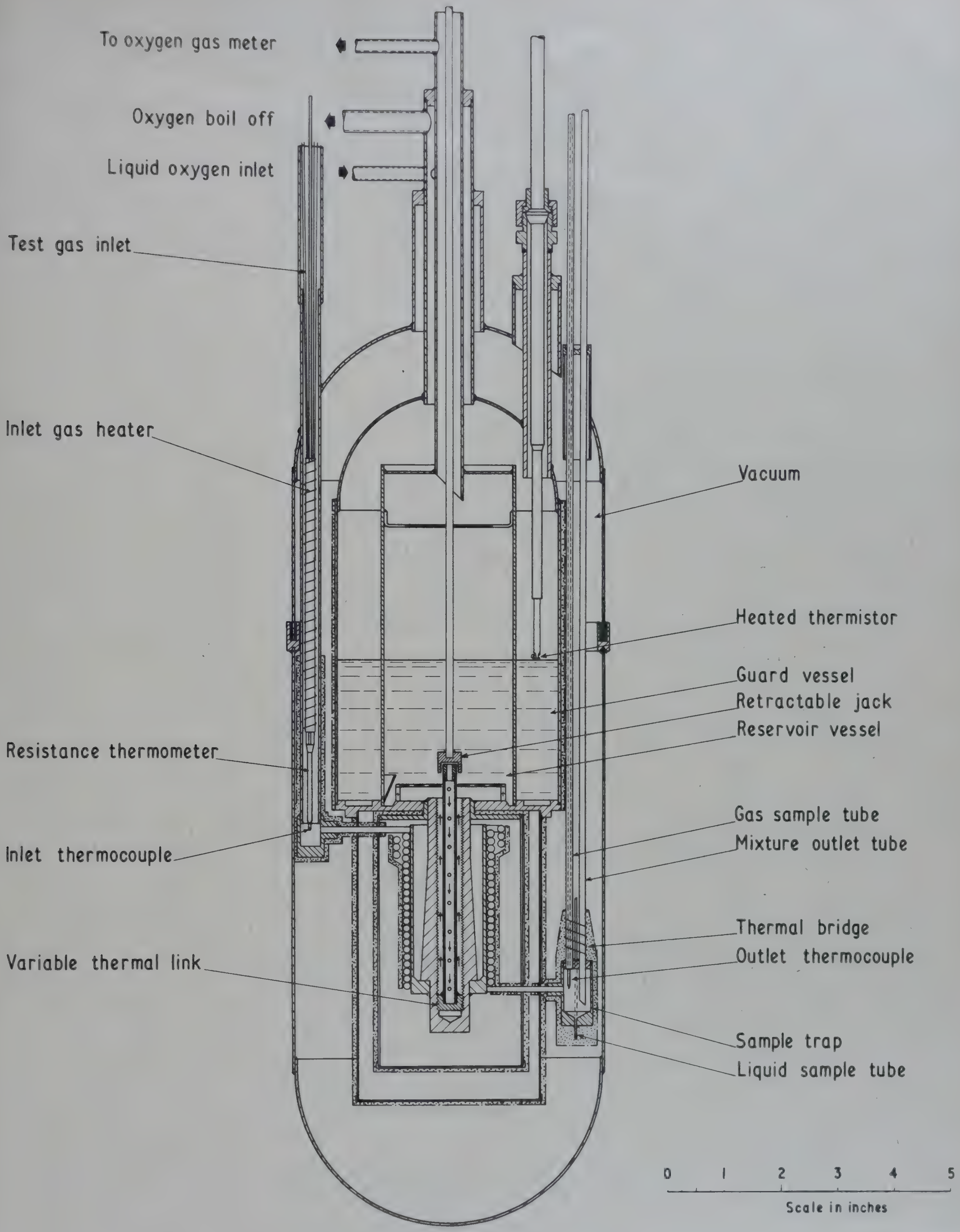
To oxygen gas meter
Oxygen boil off
Liquid oxygen inlet
Test gas inlet
Inlet gas heater
Resistance thermometer
Inlet thermocouple
Variable thermal link
Vacuum
Heated thermistor
Guard vessel
Retractable jack
Reservoir vessel
Gas sample tube
Mixture outlet tube
Thermal bridge
Outlet thermocouple
Sample trap
Liquid sample tube
0
1
2
3
4
5
Scale in inches

Fig.1. Calorimeter design

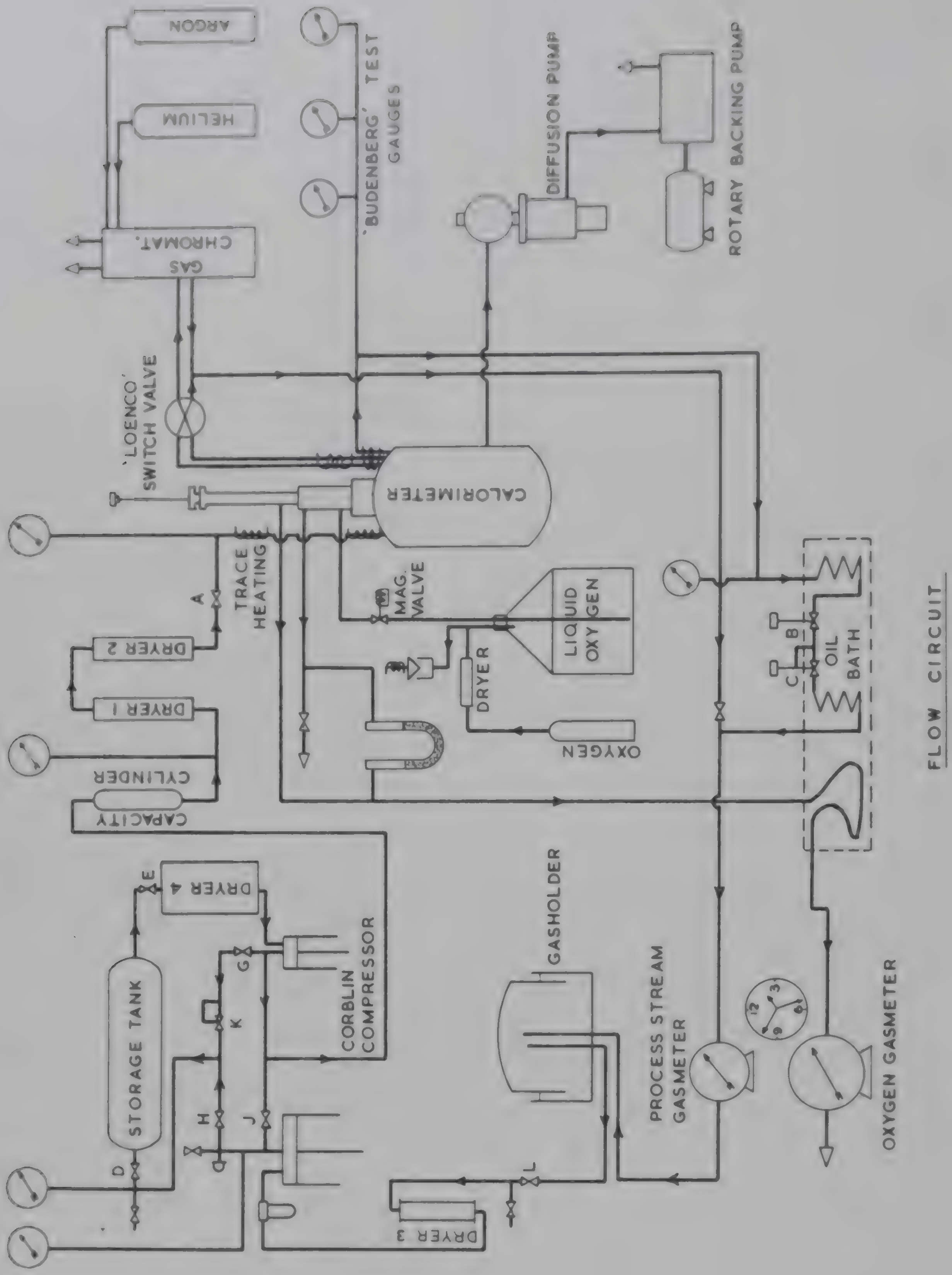
ARGON
HELIUM
GAS CHROMAT.
'BUDENBERG' TEST GAUGES
DIFFUSION PUMP
ROTARY BACKING PUMP
'LOENCO' SWITCH VALVE
CALORIMETER
TRACE HEATING
MAG. VALVE
LIQUID OXYGEN
DRYER
OXYGEN
A
DRYER 2
DRYER 1
CYLINDER
CAPACITY
B
C
OIL BATH
FLOW CIRCUIT
DRYER 4
E
STORAGE TANK
G
K
CORBLIN COMPRESSOR
H
J
D
GASHOLDER
PROCESS STREAM GASMETER
12
3
6
9
OXYGEN GASMETER
L
DRYER 3

Fig.2. Flow circuit

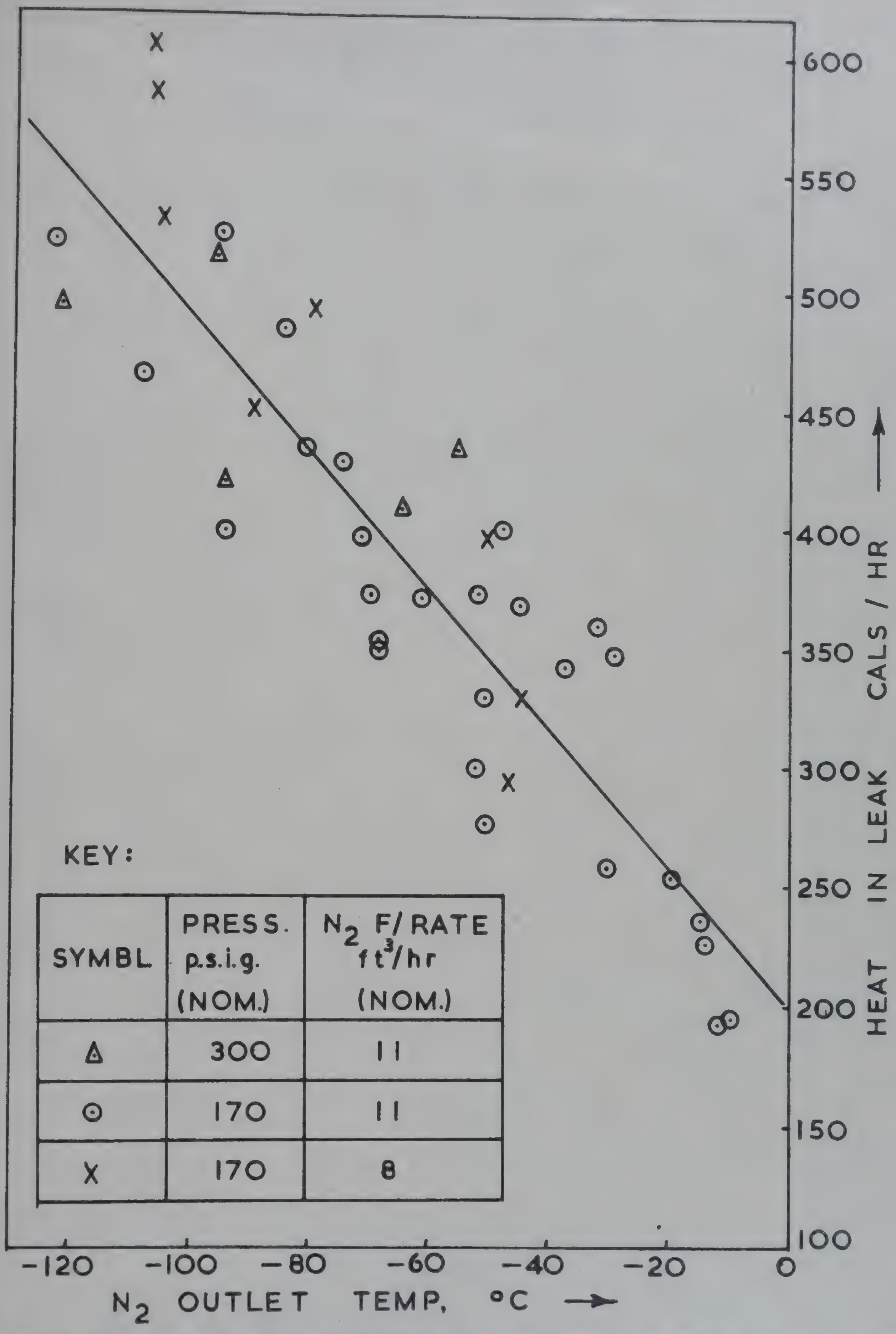

Fig.3. Heat in-leak to the calorimeter as a function of flow and temperature

STANDARDS DEVELOPMENT IN THE UNITED STATES FOR LIQUEFIED NATURAL GAS INSTALLATIONS

By A. F. Dyer* and E. C. Sommer**

Due to the rapid growth of energy demands, the transportation, storage and use of natural gas has grown by great proportions. The most economical method of transporting and storing natural gas in large quantities is in the liquid form at a low temperature (about -260 F) and pressure (generally less than 1 psig).

Liquefied natural gas, or LNG, as it is commonly called, presents an interesting contrast; on the one hand, LNG is considered by some as a relatively new product and many people feel that little is known about it; on the other hand, the rapid increase in recent years in the number of LNG storage facilities and the considerable coverage of them by the trade and technical press, symposia, etc., has resulted in such frequent use of the term "LNG" that the subject seems quite common to many of us.

While refrigerated storage of LNG is not new, problems and questions have arisen because the development of standards and guides did not take place as rapidly as did the use of LNG. Therefore, in many locations and countries, authorities have been requested to review or approve LNG installations for which they had no standards or guides. Since, in most cases, experience and technology for proper design and construction of an LNG facility was not available to these local authorities, they were understandably often at a loss as to what guidelines should be used in evaluating of safety features and design of a proposed LNG storage facility. This lack of knowledge sometimes led to misunderstanding and fear, resulting in proposed requirements which were either overly restrictive or ultra-conservative when the physical properties of LNG were considered. This has been particularly true when some of the proposed requirements for LNG facilities were compared with existing standards for other fuels such as liquefied petroleum gas, gasoline, etc.

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Until recently, this problem existed in the United States. As a result, both the American Petroleum Institute (API) and the National Fire Protection Association (NFPA) have now developed standards for the design of LNG storage facilities (API 2510A, "Design and Construction of LNG Installations at Petroleum Terminals, Natural Gas Processing Plants, Refineries, and Other Industrial Plants", and NFPA 59A, "Liquefied Natural Gas at Utility Gas Plants"). While there are some differences between these two standards, it must be clearly recognized that they cover different types of installations. NFPA Standard No. 59A is written to cover installations at utility gas plants, while API Standard 2510A covers LNG installations at petroleum terminals, natural gas processing plants, refineries and other industrial plants not covered by NFPA 59A. Although there is no specific minimum tank capacity in either standard, essentially all the requirements in both standards were written around the large volume (100,000 gallons and larger) installations.

There is activity in the United States, which will be discussed further later, to reconcile the differences between these two standards with the ultimate goal of developing a single LNG standard.

This paper will review briefly the basic requirements contained in API 2510A and NFPA 59A as well as some of the background and thinking that went into the development of the requirements of these standards.

First, let us consider some background of this problem of lack of standards. The first LNG storage facility was constructed in Cleveland, Ohio in the early 1940's. The facility, which supplied natural gas for peak shaving purposes, stored LNG at a temperature of about -250 F and a pressure of about 2 psig. The facility was operated successfully until the fall of 1944 when one of the tanks suddenly failed.

The specific details of the causative factors surrounding the failure of the tank, though interesting, are not, in themselves, significant to this discussion. One of the principal conclusions resulting from the investigation of the incident, however, was that the lack of adequate standards, including requirements for low temperature materials, in particular, for such an installation probably contributed to the accident. The consequences of the incident are important in that (1) it set the development of LNG storage facilities in the U. S. back about 20 years, and (2) it contributed to advances in the technology for and fabrication of cryogenic equipment. Much work has been done by industry since the Cleveland incident on behavior of metals at the low temperatures involved in LNG storage. The result has been tremendous advances in metallurgy and tank fabrication for service at low temperatures, such as are used in LNG storage. Today, we know that LNG tanks can be designed and constructed with the desired high degree of integrity common in the petroleum industry.

NFPA No. 59A and API 2510A both have sections devoted to construction of LNG tanks. API Standard 2510A requires that metallic tanks be fabricated in accordance with Appendix Q, "Low-Pressure Storage Tanks for Liquefied Natural Gas", of API Standard 620, "Recommended Rules for Design and Construction of Large, Welded, Low-Pressure Storage Tanks", (or equivalent). Appendix Q spells out design requireme and criteria as well as suitable metals for the service temperatures involved. In addition to referencing API Standard 620, NFPA Standard 59A also permits container designs in accordance with Section VIII of the ASME Boiler and Pressure Vessel Code provided that certain metals are used. Both the NFPA and API standards provide for the use of non-metallic materials such as concrete for tank construction, so long as the materials and the overall designs are suitable for the operating temperatures.

The basic premises underlying the safe design of any fuel storage facility, including one for LNG, are (1) that the storage tank will not be subject to complete or catastrophic failure and (2) that the remainder of the facility will be designed to provide such high integrity so as to minimize the possibility of uncontrolled release of excessive quantities of liquid or vapor. These premises have been used and proved to be valid by many years of experience with storage tanks for various fuels. The same basic premises and design principles are valid for LNG storage as has been demonstrated by the numerous facilities now in operation. It is obvious the low operating temperatures involved make it especially necessary that competent organizations and companies use sound engineering practices in applying the guidelines contained in standards such as API 2510A, NFPA 59A, API 620, and the ASME Boiler and Pressure Vessel Code, to achieve the goal of minimizing the possibility of uncontrolled release of LNG.

Standards such as API 620 and the ASME Boiler and Pressure Vessel Code provide design criteria, requirements for materials of construction that are suitable for the intended operating conditions, pressure relief valve requirements for normal and emergency conditions, etc. to ensure that the storage tank will maintain its integrity.

A question sometimes arises as to the possibility of an internal explosion. This is not a justifiable concern because, just as in the case of gasoline or refrigerated LP-gas storage tanks, the vapor space of an LNG tank is too rich to burn.

The design of the remainder of the facility to provide such high integrity so as to minimize the possibility of uncontrolled release of excessive quantities of liquid or vapor may be summarized as follows:

1. Sometime during the life of the LNG facility, a leak or spill might occur. This leak or spill should be contained and controlled.
2. LNG facilities should be located or arranged so that leaks and spills can be controlled and minimize the possibility of endangering adjacent installations.
3. LNG facilities should be located so that emergencies at adjacent installations will not seriously endanger the LNG facilities.

Let's examine these points and show how they have led to specific requirements in API 2510A and NFPA 59A.

First of all, let's assume a leak or spill does occur. How do we contain and control it?

For aboveground containers, NFPA 59A and API 2510A both provide that aboveground LNG containers be diked or drainage be provided so that accidental discharge of liquid will not endanger adjacent properties or reach waterways.

The need for such provisions is based upon generally accepted good practice, experience and tests. Tests conducted by the U. S. Bureau of Mines near Lake Charles, Louisiana and reported in their publication, "Fire and Explosion Hazards Associated with Liquefied Natural Gas", concluded that it is important to confine spills within the smallest feasible diked area and diking practices should be similar to those for gasoline storage.

The concept of diking aboveground LNG storage, or providing drainage to a safe location, is basically the same as for the storage of any liquid fuel - gasoline, jet fuel, refrigerated LP-gas, etc. One dike enclosure or drainage catch basin may serve more than one tank if the net capacity of the dike or drainage basin is at least equal to the capacity of the largest tank. These provisions have been incorporated in both NFPA 59A and API 2510A. In addition, NFPA 59A requires that the dike wall be at least five feet high while API 2510A has no minimum dike height. Both standards also require that dike walls and tank foundations be capable to withstand exposure to LNG.

Thus, with dikes, a leak or spill can be contained. Control of a leak or spill, that is shutting off the flow of LNG, requires adequate valves in the system. Both standards require shut-off valves on all tank connections below any liquid level and that these shut-off valves be located as close to the tank as practical. API 2510A provides that these valves may be automatically or remotely operated when desired. NFPA 59A requires that these valves be either automatically or remotely operated.

NFPA 59A further recommends that shut-off valves (preferably either automatically or remotely operated) also be installed in the liquid piping system at frequent intervals near tanks, buildings, etc. to provide a means of limiting the volume of liquid that could be discharged in the event of a pipe failure. API Standard 2510A does not include this specific recommendation. Instead, it considers that this type of detail should be handled in the engineering of each installation after consideration of its particular needs and requirements.

The second design premise for minimizing the effect of a leak or spill, as we previously indicated, is to locate the LNG facility so that a leak or spill would not seriously endanger adjacent installations or properties. As with diking, spacing requirements for LNG tanks were developed based on both tests and experience.

The test work done by the U. S. Bureau of Mines showed that with a spill or leak onto warm surroundings, such as into a diked area, there is an initial heat pickup which produces a large generation of vapor for a short time interval of about 1/2 to 1 minute. Thereafter, the vaporization rate due to heat pick-up is quickly reduced to a quasi-steady-state since the ground is quickly frozen and acts as an insulator. The zone of flammable vapor travel is then restricted to the immediate vicinity of the spill. In test spills at Lake Charles, Louisiana conducted in 1961, it was found that flammable concentrations of natural gas vapor could not be detected at grade immediately downwind of a 5 ft. x 5 ft. diked area containing LNG 100 seconds after the initial spill. This is quite understandable since the vapor quickly warms to the point where it is lighter than air. Therefore, a spill or leak should not offer a major concern from the standpoint of a large vapor cloud which could spread over a large area. By contrast, gasoline vapors have a density of about four times that of LNG vapor whereas LP-gas vapor has a density of three to four times that of LNG vapor. Also, the lower flammable limit of methane is about two times higher than that for LP-gas vapor. Thus, in many respects LNG spills would not offer as great a concern as should a spill of refrigerated LP-gas. These could be considered to be "safety-plus" factors for LNG.

The studies conducted by the Bureau of Mines were utilized along with comparison of the properties of methane, LP-gas and gasoline vapors plus experience in handling them in developing the spacing requirements recommended in API 2510A and NFPA 59A.

Both of these standards require that large containers be at least 200 feet from property lines that may be built upon. (API Standard 2510A recognizes that containers of less than 100,000 gallons capacity can be located much nearer property lines.) NFPA also requires that containers be at least 200 feet from public ways whereas API requires a minimum of 100 feet.

One other spacing criteria that is of interest is the location of the dike with respect to property lines. API 2510A has not considered such a restriction as necessary however, NFPA 59A requires a separation of at least 100 feet.

These distances, in some respects, are considered to be somewhat conservative when considering the nature of LNG containers as compared to LP-gas and gasoline storage tanks. For example, the typical aboveground LNG container is a double wall vessel, with several feet of space between the two shells which is filled with insulating material. This type construction provides good protection against possible fire exposure. Again, these are safety-plus factors when considering LNG storage as compared to typical LP-gas pressure storage or gasoline storage installations. Therefore, these spacing requirements should offer a more than adequate degree of protection to the LNG container when considering possible fire exposure.

The third design premise for minimizing the effect of a leak or spill, you will recall, was to locate the storage facility so that emergencies at adjacent installations or properties would not endanger the LNG facility. As one might imagine, the distances required to protect adjacent installations or properties from an emergency at the LNG facility (our second premise, just discussed) will generally be sufficient to satisfy the separation requirements of this third premise. Obviously, peculiar features of a particular location must be taken into consideration to determine if the separation requirements are adequate for the situation.

Related to the separation requirements from property lines and rights-of-way are the in-plant separation requirements between the storage tank and related process equipment. API 2510A requires that a continuous source of ignition (such as fired process equipment) be at least 100 feet from an aboveground or inground LNG storage tank. NFPA 59A requires that all fired process equipment be at least 100 feet from an inground tank, 100 feet from the dike surrounding an aboveground tank(s) and 50 feet from non-fired process equipment.

NFPA 59A requires that spacing between LNG containers be at least one-fourth the sum of their diameters while API requires a minimum of one-half the diameter of the smaller tank, but states this need not exceed 100 feet.

Fire protection is another area which deserves some discussion. Neither NFPA nor API requires fixed fire protection systems for LNG installations. However, portable dry chemical fire extinguishers and water for cooling purposes are generally provided at most facilities and meet the intent of either standard. The basic concept in both the API and NFPA standards is that a major fire should be controlled - not extinguished.

It is often proposed that dry chemical units for extinguishing any fire in a diked area be provided. This is impractical and unnecessary. The U. S. Bureau of Mines report indicates that a minimum application rate of about 0.14 pounds per second per square foot with about a one minute supply would be necessary to effect extinguishment. Assuming an area of 60 ft. x 60 ft., an application rate of about 500 pounds per second would be required. This would require about 30,000 pounds of dry chemical for a one minute application rate. Equipment to carry and apply this quantity of chemical is not practical even if good distribution of the chemical over the burning area could be achieved. This does not, however, detract from the merit of portable dry chemical extinguishers which can be used effectively to extinguish small fires.

High expansion foam has been used successfully to extinguish some test fires. However, insufficient data is not available to indicate that high expansion foam would be effective on a large LNG diked area fire.

The present fire control concept is that large spill fires should be allowed to burn themselves out while exposed facilities are kept cool by use of water. This appears to be a practical and sound approach when considering the heavy insulation used to protect LNG piping and tanks.

One other point which often comes into question is the safety relief valve protection for LNG containers. Both API and NFPA standards permit the discharge of vapor from relief valves to the atmosphere. While on initial release, LNG vapors are heavier than air, they quickly warm up and become diluted with air to below the flammable limit. Therefore, there is no danger that vapor discharges will settle back to grade to offer a fire problem. This is supported by both test work and experience.

As a result of mutual concern and desire to create a common LNG standard which could be applied to any type of LNG facility, steps are now being taken to review, refine and correlate API 2510A and NFPA 59A into a single standard. This new standard will be developed in a new NFPA Committee (tentatively named the NFPA Technical Committee on Liquefied Natural Gas). The membership of this committee will include representatives of the American Petroleum Institute, the American Gas Association, individual consultants in the field, insurance groups and others (including Federal and state regulatory personnel) having an interest in LNG storage facilities. It is planned that this new NFPA standard will ultimately be presented to the United States of America Standards Institute for adoption as a USA Standard.

This provides some of the highlights of the API 2510A and NFPA 59A standards. Time does not permit mentioning all points covered by the standards, thus, reference should be made to each standard for additional information. As noted previously, both of these standards summarize generally accepted good engineering practices that will result in the design and construction of facilities to provide a system of high integrity to minimize the possibility of uncontrolled releases of excessive quantities of liquid. This concept is not new. It has been used many times for many years and has resulted in the petroleum and gas industries' outstanding safety records. We have only applied this concept to the design, construction and installation of LNG storage facilities.

PHASE PROPERTIES OF NATURAL GAS WITH PRAUSNITZ - CHUEH METHOD

GIANFRANCO GUERRERI

ABSTRACT

For predicting the condensation of natural gas, rapid and accurate method for calculating thermodynamic properties of hydrocarbon mixtures are needed. Rigorous calculations are imperative because of the complex nature of non ideal mixtures.
The wide possibility to use digital computers makes it possible to employ sophisticated calculation procedures.

A new method has recently appeared, due to Proff. Prausnitz and Chueh, which eliminates inconveniences due to arbitrary hypothetical standard states, erroneous liquid partial molar volumes and dependence of equilibrium K constants only from pure component properties.

Besides the pure components, binary system components characteristics are taken into account which allow more accurate K values.

Comparison of K values calculated with the new method and others of more common use with experimental values gives confirmation of the above statement.

INTRODUCTION

Accurate methods for calculating the properties of hydrocarbon mixtures are needed for predicting the behaviour of natural gas processes. For example, calculation must be made for composition of equilibrium vapor and liquid phases in two phase separation of gas processing. In addition, predictions must be made for the temperature and pressure changes associated with the flow of single and two-phase natural gas fluids through heat exchangers and turbines.

The comple nature of nonideal mixtures makes rigorous calculations for these natural gas properties difficult and time consuming.

The practical method for calculating properties of natural gas mixtures without a digital computer is the use of generalized correlations. These are usually based on "average" properties of liquid and vapor mixtures and have proved to be fairly satisfactory for estimating properties of many hydrocarbon systems. Digital computers, however, have made practical the use of equations of state and more sophisticated methods for calculating the phase properties of natural gases.

It has been said that, if basic pure component data are available, all the desired thermodynamic properties of mixtures can be derived from a single equation of state describing the mixture. Potentially this is true provided reliable mixing rules are available. In addition an equation of state capable of calculating phase properties for a wide variety of vapor and liquid mixtures up to the critical point is not known (1).

A new calculation procedure follows from the convinction of Proff. Prausnitz and Chueh (2) that whereas it is extremely difficult to predict phase behaviour from pure component data alone one may expect with few exceptions to predict with confidence multicomponent behaviour from at least some binary data in addition to pure component data. Binary characteristics are obtained from careful reduction of binary experimental data. Fortunately the literature is already rich with binary data and more binary experimental results are appearing regularly.

The success of Prausnitz-Chueh method largely depends upon the fullest possible use of all the thermodynamic tools that have recently become available; among them, the new technique for estimating liquid partial molar volumes (1) is expecially important. Secondly, their method avoids arbitrary hypothetical standard states for volatile components. These are unsatisfactory because it is usually not possible to specify a consistent set of properties for those states and because there is no assurance that the desired relationship (3) for light component

$$\text{Limit } \gamma_2 = 1$$
$$\text{as } x_2 \to 1$$

will always be attained, regardless of the nature of the other components. A more satisfactory procedure is to define the reference fugacity for the light component by:

$$f_2^{\circ} = \lim_{x_2 \to 0} \frac{f_2}{x_2} = H \qquad (1)$$

The reference fugacity as defined by eq. (1) corresponds to the Henry's law constant which has the common symbol H.
The advantage of using this reference fugacity is that it is unambiguously defined, being derived from real rather than imaginary physical data: then there is no question about its numerical value for any given system.
The disadvantage of this reference fugacity is that it depends both on the properties of the light component and those of the heavier components; therefore this reference fugacity for a given

solute is also a function of the solvents.

CALCULATION METHOD

Equality of fugacities of any component "i" in vapor and liquid phases is the criterion for vapor liquid equilibrium. Vapor phase fugacity coefficient φ_i and liquid phase activity coefficient γ_i are introduced to facilitate thermodynamic analysis.

$$f_i^V = \varphi_i y_i P = f_i^L = \gamma_i x_i f_i^o \tag{2}$$

Vapor phase coefficient φ_i are calculated by using the Redlich-Kwong (4) equation of state (See appendix 1 for demonstration)

$$\ln \varphi_i = \ln \frac{v}{v-b} + \frac{b_i}{v-b} - \frac{2\sum_{j=1}^{N} y_j a_{ji}}{RT^{3/2} b} \ln \frac{v+b}{v} + \frac{a b_i}{RT^{3/2} b^2}\left[\ln \frac{v+b}{v} - \frac{b}{v+b}\right] - \ln \frac{Pv}{RT} \tag{3}$$

The liquid phase activity coefficient γ_i depends on the liquid phase composition and also on the total pressure at constant temperature. For thermodynamic analysis of high pressure vapor-liquid equilibria it is important to separate the effect of pressure from that of composition. It is then useful, for example for a binary system, to define two adjusted activity coefficients independent of pressure by (7)

$$\gamma_1^{(P_0)} = \frac{f_1}{x_1 f_{pure\,1}^{(P_0)}} \exp \int_P^{P_0} \frac{\bar{v}_1^L}{RT} \, dP \tag{4}$$

$$\gamma_2^{*(P_0)} = \frac{f_2}{x_2 H_{2(1)}^{(P_0)}} \exp \int_P^{P_0} \frac{\bar{v}_2^L}{RT} \, dP \tag{5}$$

$\gamma_1^{(P_o)}$ and $\gamma_2^{*(P_o)}$ are the liquid phase activity coefficients of respectively the less and more volatile components (1) and (2) at the reference pressure (Po). This is arbitrary and is most conveniently set equal to zero.

The asterisk (*) in equation (5) is a reminder that the unsymmetric convention has been used for normalization of activity coefficients. For subcritical component (1) the reference state is pure liquid at temperature of the solution and reference pressure (Po).

For supercritical component (2) the reference state is its infinitely diluted solution in the solvent (1) at the temperature of the solution and reference pressure (Po). This corresponds to the Henry law constant H.

As a result, both activity coefficients approach unity as the liquid solution becomes infinitely dilute with respect to the light component:

$$\gamma_1^{(P_o)} \rightarrow 1 \qquad \text{as} \quad x_1 \rightarrow 1 \tag{6}$$

$$\gamma_2^{*(P_o)} \rightarrow 1 \qquad \text{as} \quad x_2 \rightarrow 0 \tag{7}$$

The unsymmetric convenction has the advantage that it does not introduce hypothetical liquid reference state for the noncondensable supercritical component. It has been repeatedly found that for a noncondensable component, the hypothetical pure-liquid re ference state obtained from vapor-liquid equilibria of that component in various solvents may differ considerably from each other.

As shown in ref. (7) the constant pressure activity coefficients defined by eqs. (4) and (5) with the unsymmetric convenction satisfy the isothermal isobaric Gibbs-Duhem equation

$$x_1 d\ln\gamma_1^{(P_0)} + x_2 d\ln\gamma_2^{*(P_0)} = 0 \qquad (8)$$

The composition dependence of these activity coefficients can then be represented by an integrated form of eq. (8) (for example, the van Laar equation of the Margules equation) as used in low pressure vapor-liquid equilibria.

Following the development of Scatchard (8) the following equation defines the excess Gibbs energy per mole of solution.

$$\frac{g^{E*}}{RT} = x_1 \ln\gamma_1^{(P_0)} + x_2 \ln\gamma_2^{*(P_0)} \qquad (9)$$

In a manner analogous to that of Whol (9) the excess Gibbs energy can be represented by summing interaction molecules

$$\frac{g^{E*}}{RT(x_1q_1+x_2q_2)} = -\alpha_{22(1)}\,\phi_2^2 - \cdots\cdots \qquad (10)$$

where:

$$\phi_2 = \frac{x_2q_2}{x_1q_1+x_2q_2} \qquad (11)$$

Activity coefficients can be found from the exact relations:

$$R\ln\gamma_1^{(P_0)} = \left(\frac{\delta g^{E*}}{\delta n_1}\right)_{T,P,n_2} \qquad (12)$$

$$R\ln\gamma_2^{*(P_0)} = \left(\frac{\delta g^{E*}}{\delta n_2}\right)_{T,P,n_1} \qquad (13)$$

where n_1 and n_2 are the number of moles of component 1 and 2. Equations (10), (11), (12) and (13) yield the following expression for activity coefficients

$$\ln \gamma_1^{(P_0)} = A\phi_2^2 + B\phi_2^4 \tag{14}$$

$$\ln \gamma_2^{*(P_0)} = A\left(\frac{v_{c2}}{v_{c1}}\right)\left(\phi_2 - 2\phi_2\right) + B\left(\frac{v_{c2}}{v_{c1}}\right)\left(\phi_2^4 - \frac{4}{3}\phi_2^3\right) \tag{15}$$

where

$$A \equiv \alpha_{22(1)} v_{c1} \tag{16}$$

$$B \equiv 3\eta_{2(1)} \alpha_{22(1)} v_{c1} \tag{17}$$

q_1 and q_2 in equation (10) and (11) are given by

$$q_1 = v_{c1}\left(1 + \eta_{2(1)} \phi_2^2\right) \tag{18}$$

$$q_2 = v_{c2}\left(1 + \eta_{2(1)} \phi_2^2\right) \tag{19}$$

where the dilation constant $\eta_{2(1)}$ takes account of how much the light component dilates the liquid solution.

The liquid partial molar volumes $\bar{v}_1^L$ and $\bar{v}_2^L$ which appear in Eqs. (4) and (5) are calculated by the following

$$\bar{v}_i^L = \frac{\dfrac{RT}{v-b}\left(1 + \dfrac{b_i}{v-b}\right) - \dfrac{2\left(\sum_j^N x_j a_{ij}\right) - \dfrac{a b_i}{v+b}}{v(v+b)T^{1/2}}}{\dfrac{RT}{(v-b)^2} - \dfrac{a}{T^{1/2}}\left[\dfrac{2v+b}{v^2(v+b)^2}\right]} \tag{20}$$

This is obtained by the exact thermodynamic relation:

$$\bar{v}_i^L = \left(\frac{\delta V}{\delta n_i}\right)_{P,T,n_j(j \neq i)} \tag{21}$$

using the Redlich-Kwong equation of state as illustrated in Appendix 2. The mixing rules as given by Prausnitz and Chueh are somewhat different from those of the vapor phase.

The value of the liquid mixture molar volume "V" in eq. 20 is calculated by a procedure suggested by Prausnitz and Chueh (10) provided by a corresponding-states correlation specifically developed for saturated liquids.

COMPARISON OF RESULTS WITH EXPERIMENTAL AND CALCULATED DATA

An expression for the equilibrium constant K_i can be deduced by equations (2), (4) and (5)

$$K_1 = \frac{\gamma_1^{(P_0)} f_{pure\,1}^{(P_0)}}{\varphi_1 P \exp \int_P^{P_0} \frac{\bar{v}_1^L}{RT} dP} \qquad (12)$$

$$K_2 = \frac{\gamma_2^{*(P_0)} H_{2(1)}^{(P_0)}}{\varphi_2 P \exp \int_P^{P_0} \frac{\bar{v}_2^L}{RT} dP} \qquad (13)$$

respectively for less volatile and more volatile component.

Price, Kobayashi and Riki (11) published experimental results of equilibria for the system Methane-Ethane-Propane. In the following tables a comparison of K values calculated with the Prausnitz-Chueh method and others of more common use with experimental values is presented.

The alternative calculation procedures are those due to Chao and Seader (12), Lenoir (13), (14), (15). Benedict Webb Rubin (16) (17).

The temperature ranges from 172°K to 283°K.

The pressure covers the field from 6,8 to 38,4 atm.

For each K_i calculated value, the percent difference respect to the experimental K_i is calculated. In many cases the results of the Prausnitz-Chueh method are in agreement with the experimental values much better than the corresponding results of the alternative calculation methods.

This is true for the individual K_i constants and even in those cases where they are some worst than the alternative ones, the errors are distributed in such a manner that the final bubble pressure agrees better with the experimental value.

For two cases with high methane content in liquid phase, the Chao-Seader method was not able to give any result as anticipated by Lenoir (18) in his article for the limits of applicability of Chao and Seader method.

CONCLUSIONS

The Prausnitz and Chueh calculation procedure has been used as a basis for calculating vapor-liquid equilibria for mixtures of light hydrocarbons always present in natural gas. Mixtures of only three components have been considered because of the difficulty to have experimental data of low temperature - high pressure for multicomponent systems. Otherwise, natural gas is mainly composed of methane, ethane and propane and these three components govern its condensation.

Other three common calculation procedures have been considered and all the results confronted with available experimental data.

Percentage difference in K_i values and bubble pressure have been calculated and given in tables. The smallest difference in K_i values have been observed for methane that, obviously, is the most important component for natural gas condensation.

APPENDIX 1

The Redlich-Kwong equation is:

$$P = \frac{RT}{v-b} - \frac{a}{T^{0.5} v (v+b)} \tag{A-1}$$

where

$$a = \frac{\Omega_a R^2 T_{ci}^{2.5}}{P_{ci}} \tag{A-2}$$

$$b = \frac{\Omega_b R T_{ci}}{P_{ci}} \tag{A-3}$$

Prausnitz and Chueh propose to evaluate Ω_a and Ω_b for each pure component by fitting equation (A-1) to the volumetric data of the saturated vapor. A list of Ω_a and Ω_b for the saturated vapors of 19 pure substances has been given (5).

To apply Eq. (A-1) to mixture, Prausnitz and Chueh propose the following mixing rules

$$b = \sum_{i=1}^{N} y_i b_i \tag{A-4}$$

$$a = \sum_{i=1}^{N} \sum_{j=1}^{N} y_i y_j a_{ij} \qquad (a_{ij} \neq \sqrt{a_{ii} a_{jj}}\ ) \tag{A-5}$$

$$a_{ij} = \frac{(\Omega_{ai} + \Omega_{aj}) R^2 T_{cij}^{2.5}}{2 P_{cij}} \tag{A-6}$$

$$P_{cij} = \frac{Z_{cij} R T_{cij}}{v_{cij}} \tag{A-7}$$

$$v_{c_{ij}}^{1/3} = \frac{1}{2}\left(v_{c_i}^{1/3} + v_{c_j}^{1/3}\right) \qquad (A-8)$$

$$z_{c_{ij}} = 0.291 - 0.08\left(\frac{\omega_i + \omega_j}{2}\right) \qquad (A-9)$$

$$T_{c_{ij}} = \sqrt{T_{c_{ii}} \cdot T_{c_{jj}}}\,\left(1 - k_{ij}\right) \qquad (A-10)$$

The fugacity coefficient is a function of pressure, temperature and gas composition: it is related to volumetric properties by the following exact relation (6):

$$RT \ln \varphi_i = \int_v^\infty \left[\left(\frac{\partial P}{\partial n_i}\right)_{T,V,n_j} - \frac{RT}{v}\right] dv - RT \ln Z \qquad (A-11)$$

By substituting eq. (A-1) and the mixing rules, eqs. (A-4) to (A-10) into (A-11) the fugacity coefficient of component "i" in the mixture becomes:

$$\ln \varphi_i = \ln \frac{v}{v-b} + \frac{b_i}{v-b} - \frac{2\sum_{j=1}^{N} y_i a_{ij}}{RT^{3/2}\, b} \ln \frac{v+b}{v} +$$

$$+ \frac{a\, b_i}{RT^{3/2}\, b^2}\left[\ln \frac{v+b}{v} - \frac{b}{v+b}\right] - \ln \frac{Pv}{RT} \qquad (A-12)$$

APPENDIX 2

The Redlich-Kwong equation of state (A-1) is still valid for liquid mixtures provided a_{ij} is not calculated with equation (A-6) but with the following one

$$a_{ij} = \frac{\frac{1}{4}\left(\Omega a_i + \Omega a_j\right) R T_{c_{ij}}^{1.5} \left(v_{c_i} + v_{c_j}\right)}{0.291 - 0.04\left(\omega_i + \omega_j\right)} \qquad (A\text{-}13)$$

where $T_{c_{ij}}$ is given by Eq. (A-10)

In partial molar volume $\bar{v}_i^L$ can be obtained from eq. (A-1) and the mixing rules, Eqs. (A-4) and (A-5) after performing the partial differentiation indicated in Eq. (21)

$$\bar{v}_i^L = \frac{\frac{RT}{v-b}\left(1+\frac{b_i}{v-b}\right) - \frac{2\left(\sum_j^N x_j a_{ij}\right) - \frac{a b_i}{v+b}}{v(v+b)T^{1/2}}}{\frac{RT}{(v-b)^2} - \frac{a}{T^{1/2}}\left[\frac{2v+b}{v^2(v+b)^2}\right]} \qquad (A\text{-}14)$$

NOMENCLATURE

γ_2 = liquid phase activity coefficient of more volatile component 2

x_2 = liquid mole fraction of more volatile component 2

f_2^o = reference fugacity of more volatile component (2)

f_2 = fugacity of more volatile component (2)

H = Henry constant

φ_i = vapor phase fugacity coefficient of component "i"

γ_i = liquid phase activity coefficient of component "i"

P = total pressure of the system

R = universal gas constant

T = system absolute temperature

v = total molar volume of the mixture

a, b = constants in R-K equation of state

T_{ci} = critical temperature of component "i"

P_{ci} = critical pressure of component "i"

y_i, y_j = vapor mole fraction of component "i" and j

v_{ci} = critical volume of component "i"

Z_{cij} = critical compressibility factor characteristics of the i - j interaction

k_{ij} = characteristic constant for i - j interaction

$\gamma_1^{(P_o)}$ = liquid activity coefficient of less volatile component 1, in a binary system, at reference pressure (Po)

$\gamma_2^{*(P_o)}$ = liquid phase capacity coefficient of more volatile component (2), in a binary system, at reference pressure (Po).

f_i^V = fugacity of component "i" in vapor phase

f_i^L = fugacity of component "i" in liquid phase

f_i^o = fugacity of component "i" in reference state

f_1 = fugacity of less volatile component (1)

$f_{\text{pure } 1}^{(P_o)}$ = fugacity of pure component (1) in its reference state at temperature of the mixture and reference pressure (Po)

$H_{2,1}^{(P_o)}$ = Henry constant of volatile component (2) in solvent (1) at temperature of the mixture and reference pressure (Po)

$\bar{v}_i^L$ = liquid partial molar volume of component (i)

x_1 = liquid mole fraction of less volatile component 1

g^{E*} = excess Gibbs energy in unsymmetric convention

ϕ_2 = volume fraction as given by eq. (11)

q = effective molar volume

$\alpha_{22(1)}$ = self interaction constant of molecules 2 in the environment of molecules 1

$A;B$ = binary parameters given by eqs. (16), (17)

$\eta_{2(1)}$ = dilation constant of solute (2) in solvent (1)

z = compressibility factor

$\Omega_a;\Omega_b$ = dimensionless constants in Redlich-Kwong equation

ω = acentric factor

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Comp	x_i	y_i	K_i exp.	K_i Chao-Seader	K_i Lenoir	K_i B.W.R.	K_i Prausnitz	ΔK_i % Chao-Seader	ΔK_i % Lenoir	ΔK_i % B.W.R.	ΔK_i % Prausnitz
Pt. 1		T = 172,06°K			P = 6,8027 atm.						
CH_4	0,242	0,969	4,004	4,001	4,007	4,046	4,016	0,075	-0,075	-1,045	-0,300
C_2H_6	0,258	0,028	0,108	0,105	0,103	0,061	0,097	2,730	4,630	43,518	10,180
C_3H_8	0,500	0,003	0,006	0,0094	0,0069	0,0017	0,006	-56,670	-15,000	71,667	0,000
P			6,8027	6,029	5,090	7,678	6,599				
ΔP%			0,00	11,376	25,177	-12,867	3,000				
Pt. 2		T = 172,06°K			P = 13,605 atm.						
CH_4	0,528	0,974	1,845	1,856	1,854	1,866	1,852	-0,600	-0,488	-1,138	-0,400
C_2H_6	0,319	0,024	0,075	0,060	0,064	0,045	0,066	20,000	14,667	40,000	12,000
C_3H_8	0,153	0,002	0,013	0,0065	0,0053	0,0019	0,0065	50,000	59,231	85,385	50,000
P			13,605	14,316	12,194	14,355	12,993				
ΔP%			0,00	-5,222	10,371	-5,513	4,500				
Pt. 3		T = 199,83°K			P = 6,803 atm.						
CH_4	0,110	0,865	7,864	7,594	7,722	7,991	7,854	3,434	1,806	-1,615	0,123
C_2H_6	0,320	0,110	0,344	0,421	0,394	0,329	0,359	-22,383	-14,535	4,360	-4,360
C_3H_8	0,570	0,025	0,044	0,0525	0,043	0,026	0,037	-19,318	2,273	40,909	15,910
P			6,803	5,376	5,960	6,803	7,007				
ΔP%			0,00	20,967	12,388	0,00	-3,000				

Comp.	X_i	Y_i	K_i exp.	K_i Chao-Seader	K_i Lenoir	K_i B.W.R.	K_i Prausnitz	$\Delta K_i\%$ Chao-Seader	$\Delta K_i\%$ Lenoir	$\Delta K_i\%$ B.W.R.	$\Delta K_i\%$ Prausnitz
Pt. 4		T = 199,83°K			P = 13,605 atm.						
CH_4	0,236	0,893	3,784	3,735	3,722	3,853	3,775	1,295	1,638	-1,823	0,238
C_2H_6	0,483	0,098	0,203	0,227	0,234	0,196	0,211	-11,822	-15,271	3,448	-3,940
C_3H_8	0,281	0,009	0,032	0,031	0,030	0,018	0,025	3,125	6,250	43,750	21,875
P			13,605	11,317	13,043	13,605	13,741				
Δ P%			0,00	16,816	4,131	0,00	-1,00				
Pt. 5		T = 199,83°K			P = 27,211 atm.						
CH_4	0,496	0,956	1,927	1,931	1,914	1,917	1,917	-0,207	0,675	0,519	0,519
C_2H_6	0,242	0,036	0,149	0,148	0,176	0,147	0,173	0,672	-18,121	1,342	-16,107
C_3H_8	0,262	0,007	0,027	0,025	0,030	0,019	0,023	7,408	-11,111	29,630	14,815
P			27,211	26,028	26,907	28,211	27,211				
Δ P%			0,00	4,346	1,117	-3,675	0,00				
Pt. 6		T = 227,61°K			P = 13,605 atm.						
CH_4	0,125	0,804	6,432	6,112	6,244	6,356	6,384	4,976	2,923	1,182	0,747
C_2H_6	0,268	0,140	0,522	0,631	0,580	0,533	0,552	-20,881	-11,111	-2,107	-5,747
C_3H_8	0,607	0,056	0,092	0,110	0,106	0,086	0,089	-19,565	-15,217	6,522	3,261
P			13,605	10,460	12,428	12,855	13,741				
Δ P%			0,00	23,118	8,651	5,513	-1,00				

Comp.	x_i	y_i	K_i exp.	K_i Chao-Seader	K_i Lenoir	K_i B.W.R	K_i Prausnitz	ΔK_i % Chao-Seader	ΔK_i % Lenoir	ΔK_i % B.W.R	ΔK_i % Prausnitz
Pt. 7		T = 227,61°K			P = 27,211 atm.						
CH_4	0,277	0,799	2,884	2,888	2,854	2,879	2,892	-0,138	1,040	0,173	-0,277
C_2H_6	0,523	0,138	0,359	0,355	0,368	0,353	0,354	1,115	-2,507	1,671	1,393
C_3H_8	0,200	0,013	0,065	0,073	0,084	0,072	0,070	-12,307	-29,231	-10,769	-7,692
P			27,211	24,335	27,266	27,211	27,755				
Δ P%			0,00	10,569	-0,202	0,00	-2,00				
Pt. 8		T = 227,61°K			P = 54,422 atm.						
CH_4	0,598	0,928	1,552	1,565	1,528	1,567	1,538	-0,840	1,546	-0,966	0,902
C_2H_6	0,109	0,036	0,330	0,319	0,390	0,335	0,440	3,330	-18,182	-1,515	-33,33
C_3H_8	0,293	0,036	0,123	0,0998	0,149	0,103	0,109	18,900	-21,138	16,260	11,4
P			54,422	61,731	57,076	55,422	55,510				
Δ P%			0,00	-13,430	-4,877	-1,837	-2,00				
Pt. 9		T = 227,61°K			P = 68,027 atm.						
CH_4	0,766	0,903	1,179		1,137	1,201	1,166		-0,678	-1,866	1,100
C_2H_6	0,089	0,050	0,562		0,531	0,526	0,854		5,516	6,406	-51,96
C_3H_8	0,145	0,047	0,324		0,301	0,284	0,214		7,099	12,346	33,95
P			68,027		66,350	68,027	65,850				
Δ P%			0,00		2,465	0,00	3,2				

Comp.	x_i	y_i	K_i exp.	K_i Chao-Seader	K_i Lenoir	K_i B.W.R.	K_i Prausnitz.	ΔK_i% Chao Seader	ΔK_i% Lenoir	ΔK_i% B.W.R.	ΔK_i% Prausnitz
Pt. 10		T = 255,39°K			P = 27,211 atm.						
CH_4	0,136	0,487	3,581	3,563	3,507	3,578	3,618	0,500	2,066	0,084	-1,030
C_2H_6	0,724	0,485	0,670	0,676	0,683	0,675	0,666	-0,890	-1,940	-0,746	0,600
C_3H_8	0,139	0,028	0,201	0,180	0,200	0,200	0,187	20,450	0,497	0,497	6,970
P			27,211	26,022	27,467	27,211	27,687				
Δ P%			0,00	4,370	-0,941	0,00	-1,750				
Pt. 11		T = 255,39°K			P = 54,422 atm.						
CH_4	0,391	0,824	2,107	2,157	2,087	2,096	2,100	-2,370	0,949	0,522	0,330
C_2H_6	0,188	0,094	0,500	0,491	0,531	0,513	0,558	1,800	-6,200	-2,600	11,600
C_3H_8	0,421	0,082	0,195	0,152	0,1997	0,182	0,178	22,05	-2,410	6,667	8,700
P			54,422	52,408	53,880	53,422	54,762				
ΔP%			0,00	3,701	0,996	1,837	-0,625				
Pt. 12		T = 255,39°K			P = 74,830 atm.						
CH_4	0,585	0,838	1,432	1,514	1,480	1,458	1,434	-5,730	-3,352	-1,816	-0,140
C_2H_6	0,107	0,063	0,589	0,487	0,527	0,587	0,701	17,320	10,526	0,339	-19,010
C_3H_8	0,308	0,099	0,321	0,202	0,253	0,302	0,279	37,070	21,184	5,919	9,97
P			74,830	107,816	68,521	74,830	73,401				
ΔP%			0,00	-44,081	8,431	0,00	1,910				

Comp	x_i	y_i	K_i exp	K_i Chao-Seader	K_i Lenoir	K_i B.W.R	K_i Prausnitz	ΔK_i % Chao-Seader	ΔK_i % Lenoir	ΔK_i % B.W.R	ΔK_i % Prausnitz
Pt. 13		T = 255,39°K			P = 88,435 atm.						
CH_4	0,710	0,829	1,168		1,287		1,220		-10,188		-4,450
C_2H_6	0,030	0,025	0,833		0,518		0,900		37,815		-8,040
C_3H_8	0,260	0,146	0,561		0,272		0,411		51,515		26,700
P			88,435		77,792		83,061				
Δ P%			0,00		12,035		6,077				
Pt. 14		T = 283,17°K			P = 13,605 atm.						
CH_4	0,027	0,275	10,185	3,317	9,303	9,452	10,296	13,430	8,611	7,197	-1,100
C_2H_6	0,160	0,284	1,775	1,930	1,909	1,822	1,812	-8,730	-7,549	-2,648	-2,080
C_3H_8	0,811	0,441	0,544	0,556	0,544	0,557	0,533	-2,200	0,000	-2,390	2,020
P			13,605	12,542	13,064	13,355	13,605				
Δ P%			0,00	7,815	3,976	1,837	0,00				
Pt. 15		T = 283,17°K			P = 27,211 atm.						
CH_4	0,110	0,557	5,064	4,917	5,061	4,940	5,145	2,900	0,059	2,449	-1,600
C_2H_6	0,175	0,193	1,103	1,161	1,113	1,088	1,074	-5,260	-0,907	1,350	2,630
C_3H_8	0,715	0,250	0,350	0,358	0,347	0,366	0,344	-2,280	0,357	-4,571	1,730
P			27,211	24,002	24,033	26,211	27,279				
Δ P%			0,00	11,793	11,679	3,675	-0,250				

Comp.	X_i	y_i	K_i exp.	K_i Chao Seader	K_i Lenoir	K_i B.W.R.	K_i Prausnitz	$\Delta K_i\%$ Chao Seader	$\Delta K_i\%$ Lenoir	$\Delta K_i\%$ B.W.R.	$\Delta K_i\%$ Prausnitz
Pt. 16		T = 283,17°K			P = 54,422 atm.						
CH_4	0,286	0,722	2,524	2,702	2,524	2,520	2,545	-7,0	0,000	0,158	-0,830
C_2H_6	0,121	0,091	0,752	0,707	0,769	0,751	0,768	+6,0	-2,260	0,133	-2,130
C_3H_8	0,593	0,187	0,315	0,239	0,312	0,314	0,302	+24,1	0,952	0,317	4,130
P			54,422	52,720	52,656	52,422	54,490				
ΔP%			0,00	3,12	3,245	3,675	-0,125				
Pt. 17		T = 283,17°K			P = 68,027 atm.						
CH_4	0,389	0,707	1,817	1,467	1,858	1,844	1,838	19,300	-2,256	-1,486	-1,150
C_2H_6	0,169	0,126	0,745	0,929	0,723	0,731	0,763	-24,700	2,953	1,879	-2,420
C_3H_8	0,442	0,167	0,378	0,617	0,351	0,377	0,353	-63,230	7,143	0,264	6,600
P			68,027	89,872	63,394	68,027	66,326				
ΔP%			0,00	-32,111	6,810	0,00	2,500				

Comp	x_i	y_i	K_i exp	K_i Chao Seader	K_i Lenoir	K_i B.W.R	K_i Prausnitz	ΔK_i % Chao Seader	ΔK_i % Lenoir	ΔK_i % B.W.R	ΔK_i % Prausnitz
Pt. 18		T = 283,17°K			P = 31,633 atr.						
CH_4	0,500	0,756	1,512	1,415	1,624	1,538	1,524	6,400	-7,407	-1,719	-0,795
C_2H_6	0,037	0,030	0,811	0,863	0,678	0,752	0,811	-6,400	16,399	7,275	0,000
C_3H_8	0,463	0,214	0,462	0,563	0,351	0,451	0,451	-21,86	24,026	2,381	2,381
P			31,633	138,283	74,001	31,633	33,333				
Δ P%			0,00	-69,397	9,349	0,00	-2,083				

Notation for the above tables:

Definitions of percentage errors: $\Delta K\% = \frac{Kexp. - K}{Kexp.} \times 100$ $\Delta P\% = \frac{Pexp. - P}{Pexp.} \times 100$

THE THERMAL CONDUCTIVITY OF METHANE*

By N. Mani+ and J. E. S. Venart^x

ABSTRACT

The available data for the thermal conductivity of METHANE between 90° and 750°K at pressures from atmospheric to 1200 bars are reviewed and a least square analysis using orthogonal polynomials is performed in the liquid, vapour and supercritical states in order to present a table of best values for the pure fluid. In all three states, the thermal conductivity, λ, could be represented as a function of reduced pressure and reduced temperature. The existing data for 1 bar pressure and extrapolated high pressure λ values were correlated as a simple function of reduced temperature. Using these correlations the thermal conductivity of Methane can be obtained to within 8% from values of pressure and temperature only.

Preliminary measurements have been made using a transient hot wire apparatus in the region where no adequate data were available, namely between 200°K and 300°k at pressures from 200 bars to 600 bars. Only the results for 291°K and 258°K are presented here. The preliminary experimental results of this investigation seem to deviate by +3% from the correlating equation and serve as an indication of the need for further measurement.

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1.0 INTRODUCTION

The thermal conductivity of Methane has been a subject of investigation by many workers over the past thirty years *[1-11]. Although this property is necessary for the accurate calculation of Nusslet numbers in order to evaluate local and overall heat transfer for pipe flow, distallation column, and liquefying processes work little to date has been reported on the overall effects of pressure and temperature on thermal conductivity and much less to present a critical review of the data which exists.

Table [1] lists in chronological order, the obvious experimental work that have been carried out in measuring the thermal conductivity of methane to this date. An examination of this literature shows that although several investigators have reported thermal conductivity data, not many have covered a very wide range of temperature and pressure, further much of the existing data has not been confirmed by second determinations. Even in cases where second determinations have been carried out large deviations between the results of different investigators, are apparent.

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* Numbers in parenthesis refer to references listed at the back of the paper.

The object of this paper is to extend the knowledge of the thermal conductivity of methane in the fluid state by obtaining reliable correlations for the existing data in the liquid, vapour, supercritical regions and presenting this data in the form of a table and also in graphical form as smoothed isobars. In addition some preliminary experimental results, obtained by means of an absolute transient hot wire apparatus, are presented. These results, due to the rather poor agreement with the correlating equations generated emphasise the need for further measurement.

2.0 Correlation

As mentioned previously Table [1] lists a survey of the experimental observations made so far. With this available body of data for the thermal conductivity of methane, a least square analysis using "orthogonal polynomial approximations" [16] has been performed in the liquid ,supercritical, and vapour states. In addition a correlation of the atmospheric pressure values has also been obtained.

2.1 Correlation for Atmospheric Pressure

The available atmospheric pressure data [2,4,6,7,8,10] and also extrapolated atmospheric pressure data [3,9,10] were correlated by the following equation over the temperature range 96-973°K:

$$\lambda = \sum_{n=0}^{3} a_n T_r^{\,n} \qquad [2\cdot1\cdot1]$$

where $T_r = (T-T_c)/T_c$,

and the coefficients are

$a_0 = 20.29$

$a_1 = 21.353$

$a_2 = 6.0425$

$a_3 = -0.69281$

The agreement between the correlating function [2.1.1] and the available data is shown in Fig.[1]. While the agreement of the majority of the results is within 3%, the value reported by Johnston and Grilly [2] at 96.76°K would seem to deviate by -10% from the present correlation.

2.2 Correlation for Liquid Region

Only the experimental data reported by Ikenberry and Rice [5] and those reported by IGT [11] in graphical form were available for the temperature range 95 to 175°K. As these data were found to have discrepancies as high as 10 to 14% (possibly due to the 5 to 10% possible error in reading the values from the IGT graph) it was decided that only the results of [5] should be used to correlate the thermal conductivity of the liquid as a function of $(p-p_{sat})$ and temperature, T, in the region bounded by the conditions

$p=p_{sat}$ $\qquad$ T=175°K

T=95°K $\qquad$ p=as shown in Fig.[5]

that is

$$\lambda = f\{(p-p_{sat}), T\}$$

or

$$\lambda = \phi\left\{\left(\frac{p-p_{sat}}{p_c}\right), \frac{T}{T_c}\right\}$$

$$= \phi\{p_r, T_r\} \quad \text{———————} (2.2.1)$$

where $p_r = \dfrac{(p-p_{sat})}{p_c}$ and $T_r = \dfrac{T}{T_c}$

With the pressure expressed as a power series in p_r; equation (2.2.1) can be written as

$$\lambda = \{a_0 + a_1 p_r + a_2 p_r^2 + \cdots\cdots\cdots\} \text{ ——— } (2.2.2)$$

where a_0, a_1, a_2 .... are functions of T_r.

The coefficients a_0, a_1, a_2..... can now be expressed as polynomials in T_r, so that

$$a_0 = \sum_{n=0}^{N_0} a_{on} T_r^n \quad , \qquad a_1 = \sum_{n=0}^{N_1} a_{1n} \cdot T_r^n$$

where N_0, N_1 .. are the maximum order of the polynomials in T_r, that have been found suitable for each of the coefficients a_0, a_1...

Combining equations (2.2.2) and (2.2.3)

$$\lambda = \sum_{n=0}^{N_0} a_{on} T_r^n + \left\{ \sum_{n=0}^{N_1} a_{1n} T_r^n \right\} p_r + \left\{ \sum_{n=0}^{N_2} a_{2n} T_r^n \right\} p_r^2 + \cdots\cdots\cdots \left\{ \sum_{n=0}^{N_m} a_{mn} T_r^n \right\} p_r^M \text{———} [2 \cdot 2 \cdot 4]$$

where M = maximum order of the polynomial in p_r that has been found suitable.

The resulting least square's analysis yielded:

$$\left[\lambda \times 10^3\right] = \sum_{n=0}^{3} a_{on} T_r^n + \left\{ \sum_{n=0}^{3} a_{1n} T_r^n \right\} p_r + \left\{ \sum_{n=0}^{3} a_{2n} T_r^n \right\} p_r^2 \text{—} [2 \cdot 2 \cdot 5]$$

with

$$\left[\lambda_{sat} \times 10^3\right] = \sum_{n=0}^{3} a_{on} T_r^n \text{————} [2 \cdot 2 \cdot 6]$$

and

$a_{00} = 456.7$ $a_{10} = -94.13$ $a_{20} = 32.193$

$a_{01} = -679.76$ $a_{11} = 377.74$ $a_{21} = -125.77$

$a_{02} = 464.52$ $a_{12} = -478.48$ $a_{22} = 162.18$

$a_{03} = -164.42$ $a_{13} = 206.75$ $a_{23} = -69.396$

The agreement between the correlating function (2.2.5) and the available experimental data is shown in Fig.[2]; while the results of Ikenberry and Rice agree within 7%, the results reported by IGT were found to deviate by ±14%. The λ_{sat} values reported by Borovik et al. [1] at pressures slightly in excess of the saturation pressure were fitted to better than ±6% with the equation (2.2.6)

2.3 Correlation for Supercritical Region

Though several investigators [5 to 10] have reported results in the region there exists a band, notably in the area where λ shows a pronounced minimum(see figure [5]) where the reported measurements are scanty.

Following a similar proceedure, as outlined in the preceeding section, a least squares correlating equation was obtained from the reported data in the following form: (for the region (2) in fig.[5])

$$[\lambda \times 10^3] = \sum_{n=0}^{3} a_{0n} T_r^n + \left\{\sum_{n=0}^{3} a_{1n} T_r^n\right\} p_r + \left\{\sum_{n=0}^{3} a_{2n} T_r^n\right\} p_r^2 + \left\{\sum_{n=0}^{3} a_{3n} T_r^n\right\} p_r^3 \quad -[2.3.1]$$

where	$a_{00} = 76.669$	$a_{10} = 13.494$	$a_{20} = -1.0226$	$a_{30} = 0.039632$
	$a_{01} = -196.27$	$a_{11} = 14.909$	$a_{21} = -1.8639$	$a_{31} = 0.061606$
	$a_{02} = 260.53$	$a_{12} = -47.302$	$a_{22} = 6.0352$	$a_{32} = -0.23604$
	$a_{03} = -91.496$	$a_{13} = 22.539$	$a_{23} = -3.0212$	$a_{33} = 0.12477$

and

$$T_r = \frac{(T - T_c)}{T_c}, \qquad p_r = \frac{(p - p_c)}{p_c}$$

The deviation of the experimental results from the correlation is shown in Fig.[3]. Although the maximum deviation is ±8% this occurs in the region where λ experiences a minimum and also where the carry-over of anomalous behavior could possibly be expected with pressure near p_c.

2.4 Correlation for Vapour Region

Utilizing the proceedures of the previous two sections a correlation was next prepared for the vapour region (region (3) in Fig.[5]). The obtained relation for the data of [6 to 10] was:

$$[\lambda \times 10^3] = \sum_{n=0}^{4} a_{0n} T_r^n + \left\{\sum_{n=0}^{3} a_{1n} T_r^n\right\} p_r^{\frac{1}{2}} + \left\{\sum_{n=0}^{3} a_{2n} T_r^n\right\} p_r^1 + \left\{\sum_{n=0}^{3} a_{3n} T_r^n\right\} p_r^{\frac{3}{2}} \quad -[2.4.1]$$

where $a_{00} = 28.737$ $a_{10} = -30.321$ $a_{20} = 38.527$ $a_{30} = -6.1862$
$a_{01} = -5.6841$ $a_{11} = 44.968$ $a_{21} = -56.302$ $a_{31} = 10.448$
$a_{02} = 13.149$ $a_{12} = -23.564$ $a_{22} = 29.527$ $a_{32} = -5.7105$
$a_{03} = -0.089241$ $a_{13} = 4.063$ $a_{23} = -5.0937$ $a_{33} = 0.99663$
$a_{04} = -0.50674$

and

$$T_r = \frac{(T-T_c)}{T_c} \quad , \quad p_r = \frac{(p-p_c)}{p_c}$$

The deviations of the reported experimental results from this correlation are shown in Fig.[4]. Again the maximum deviation is ±8% which occurs at the higher pressures. Ideally a correlation based on density would be preferable here - unfortunately the inadequate PVT data available to the authors precluded any such attempt.

2.5 Table of "Best" Values

The regions in which the above equations were used to calculate the data in Table [2] are shown in Fig.[5]. In certain areas of overlap either of two equations may be used to calculate λ without deviating from the tabulated data by more than 1 percent. On the basis of the correlations presented and the deviations of the reported experimental data it would be unwise to expect the values of λ to be more accurate than ±8% over the complete field of interest until further measurements are forthcoming.

3.0 Preliminary Measurements

A transient hot wire method [13,17] was used for the absolute determination of the thermal conductivity of methane. With this method heat is generated at a known rate in an electrically heated fine platinum wire immersed vertically in the fluid under consideration, which is contained in a cylindrical enclosure initially in thermal equilibrium. The apparatus to a first approximation simulates the behaviour of an infinite line source of constant heat generation in an extended medium. The solution to the Fourier equation of heat conduction using the initial boundary conditions of constant temperature is for this situation [12,a]

$$T_{\infty}(r,t) = -\frac{q_l}{4\pi\lambda}\left\{ E_i\left(-\frac{r^2}{4\alpha t}\right)\right\} \text{———} [3.0.1]$$

At a fixed radial distance, say at the surface of the wire where r = a, the time dependance of temperature can be obtained from equation (3.0.1) by expanding the exponential integral for small values (<0.02) of $\frac{a^2}{4\alpha t}$. The result is

$$T_{\infty}(a,t) = \frac{q_l}{4\pi\lambda}\left\{ \ln(t) + \ln\left(\frac{4\alpha}{a^2\gamma}\right) + \frac{a^2}{4\alpha t} + \cdots \right\} - [3.0.2]$$

Provided that $\frac{a^2}{4\alpha t}$ is sufficiently small in the range t_1 to t_2 then

$$T_{\infty}(a,t_2) - T_{\infty}(a,t_1) \simeq \frac{q_l}{4\pi\lambda} \cdot \ln\left(\frac{t_2}{t_1}\right) \text{———} [3.0.3]$$

A plot of temperature against log (time) therefore enables the thermal conductivity to be obtained.

In an actual experimental hot wire cell the heating wire has a finite diameter (finite heat capacity) and finite length and in addition the medium is bounded. A critical examination of the limitations and necessary modifications of equation (3.0.2) when applied to an experimental cell, has been done by J.K. Horrocks and E. McLaughlin [13]. In the present apparatus modification need to be made only for the finite heat capacity of the wire, which changes the inner boundary condition for the solution of the Fourier equation of conduction and introduces into the solution, as given by equation (3.0.2) a term dependent on the heat capacities of the wire and the medium. Thus the modified solution is [12,b]

$$T_{\infty}(a,t) = \frac{q_l}{4\pi\lambda}\left[\ln(t) + \ln\left(\frac{4\alpha}{a^2\gamma}\right) + \frac{a^2}{2\alpha t} + \left\{\frac{(\omega-2)a^2}{2\omega\alpha t}\ln\left(\frac{4\alpha t}{a^2}\right)\right\}\right]$$

$$\text{———} [3.0.4]$$

The wire is assumed to have no contact resistance with the medium and to be perfectly conducting with a constant temperature over its cross-section.

3.1 Experimental Apparatus

Figure [6] shows the stainless steel pressure vessel used in the present investigation. This hot wire conductivity cell was located inside the pressure vessel, at the end of the mandrel. A wire, made of 0.0025 inch diameter platinum, with potential leads made of 0.001 inch diameter platinum wire was arranged vertically as shown in the figure. All electrical connections were taken out of the pressure vessel through the mandrel with a "Conax" teflon sealing gland.

The pressure vessel was immersed inside a thermostat bath, filled with Di-methyl-butane (F.P. = -168°C, B.P. = 28°C). This fluid was cooled by passing liquid nitrogen through the cooling coil. The temperature of the apparatus was measured accurately to hundredths of a degree with the help of a calibrated quartz thermometer. The rate of flow of liquid nitrogen through the thermostat could be maintained constant and the final temperature controlled by adjusting the power to the heating coil. When measurements were being made the control was carried out manually enabling the temperature of the bomb to be controlled within few hundredths of a degree.

Figure [7] shows the details of whole high pressure system, consisting of two parts, one containing the experimental fluid and the other containing oil, both being separated by a differential pressure null detector. Pressure was applied to the test fluid by means of a "Ruska" positive displacement pump and the gas pressure balanced by applying pressure to the oil by means of a standard "Budenberg" dead weight tester. The balance point was indicated by a differential pressure null detector, which had an accuracy of 0.01 psi ΔP, and a sensitivity varying from 2×10^{-4} psi ΔP per meter division to 0.01 psi ΔP per meter division.

3.2 Temperature Measurement Technique

Measurement of varying temperature of the hot wire as a function of time constitutes the major experimental problem in unsteady state thermal conductivity measurements. The most convenient parameter to measure is the wire resistance since from equation (3.0.3),

$$\lambda = \frac{q_L}{4\pi} \cdot \frac{\Delta \ln(t)}{\Delta R} \cdot \left(\frac{\Delta R}{\Delta T}\right) \text{———} [3 \cdot 2 \cdot 1]$$

or for small values of temperature rise, ΔT, $\left(\frac{\Delta R}{\Delta T}\right)$ can be approximated by $\left(\frac{dR}{dT}\right)$ which can be considered effectively as a constant during each measurement. From Ohm's law,

$$\Delta R = \frac{\Delta V}{I} - \frac{V}{I^2} \cdot \Delta I$$

Thus the measurement of temperature rise requires the measurement of the current and voltage transients in and across the cell wire.

In the present investigation, the current transient in the wire was avoided by using a constant current D.C. power supply (HP) with the result, $\Delta R = \Delta V/I$, and

$$\lambda = \frac{q_L}{4\pi} \; \frac{\Delta \ln(t)}{(\Delta V/I)} \cdot \left(\frac{dR}{dT}\right)$$

$$= \frac{V.I^2}{4\pi l} \cdot \frac{\Delta \ln(t)}{\Delta V} \cdot \left(\frac{dR}{dT}\right) \text{———} [3 \cdot 2 \cdot 2]$$

Figure [8] is a schematic layout of the electrical circuit used with the hot wire cell. The switching of the relay (A,B,C) was preceeded by a manual operating switch. The circuits of the relay switch were so arranged such that when the manual operating switch was kept in 'off' position, the relay switch was closed in the direction 'A1-B3-C5' and allowed current to flow from the constant current source, through the ballast resistor circuit. When the manual operating switch was in the 'on' position the relay switch was closed in the direction 'A2-B4-C6', allowing the current to flow through the 'hot wire' circuit. Prior to initiating an actual measurement current was maintained in the ballast resistor circuit with the ballast resistance carefully matched to the cell resistance. The potentiometer P1 and P2 were balanced with the potential drops across ballast resistor and standard resistor respectively. In switching on the manual operating switch a fast response relay was activated and the hot wire circuit was closed before breaking the ballast resistor circuit.

Once current was established in the hot wire circuit, the resistance of the hot wire continuously changed due to the temperature rise of the wire resulting from the dissipation of electrical energy through the wire. The out-of-balance transient potential across the cell resistance was measured as a function of time utilizing a HP-2402A Digital Voltimeter triggered externally to take measurements at the rate of 20 readings/sec. The measured values were printed out at this rate by a HP-5050B Digital recorder connected to an external clock.

3.3 Experimental Results

Only preliminary measurements, made along 2 isotherms, 291 and 258^{O}K, at four different pressure levels are reported. Each value obtained is the mean of four individual determinations taken at an interval of 30 minutes between readings. It is expected that the present results are accurate to ± 3%; more accurate determinations will be made utilizing a finer wire in the near future.

In order to assess the reliability of the method, measurements were also preformed on toluene. The results obtained agreed, within experimental accuracy, with previous measurements reported for this fluid (13,14,15).

4.0 Conclusions

4.1 The available data for the thermal conductivity of Methane has been correlated utilizing a least square orthogonal polynomial approximation. The table of "best" values has been calculated from the correlating equations which represent the available data to ±8%.

4.2 Preliminary measurements of the thermal conductivity of Methane have been obtained utilizing a transient hot wire apparatus. The results, due to their deviation from the correlation, point out the need of further measurement.

5.0 References

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6.0 Nomenclature

a = radius of the wire (m)

C_p = specific heat at constant pressure for methane (watts/kg.sec.°C)

d = differential operator

E_i ( ) = exponential integral

I = constant current (amp)

l = length of the wire (m)

p = pressure (bars)

p_{sat} = saturation pressure (bars)

p_c = critical pressure (=46.4 bars for methane)

p_r = reduced pressure

q_l = heat generated per unit length of wire (watts/m)

R = resistance of the wire (ohm)

ΔR = change in resistance (ohm)

r = radius (m)

t = time (sec)

T = temperature (°K)

T_c = critical temperature (°K) (=191.1°K for methane)

T_r = reduced temperature

ΔT = change in temperature (°K)

Greek Letters:

λ = thermal conductivity coefficient (watts/m°C)

ρ = density (kg/m^3

α = thermal diffusivity (m^2/sec) = $\lambda/\rho \cdot C_p$

$\omega = 2\rho_m \cdot C_{pm}/\rho_w \cdot C_{pw}$

γ = Euler's constant

Σ = summation parameter

Subscripts:

exp - experimental

cor - correlated

m - medium (methane)

sat - saturation

w - wire

Reference No.	Author(s)	Method Adopted	Temperature Range in °k	Pressure Range in Bars
1	Borovik, Ye. Matveyev, A. and Panina, Ye. (1940)	Steady state concentric cylinder method	103.26 112.56 145.36 172.86	Slightly in excess of the respective vapour pressures
2	Johnston, H.L. and Grilly, E.R. (1946)	Steady state concentric cylinder method	96 to 385	Atmospheric pressure.
3	Keyes, F. G. (1955)	Steady state concentric cylinder	120.45 171.05 273.15	1.1 1.2 to 6.5 1.2 to 9.0
4	Geier, Von. H. and Schafer, K. (1961)	Steady state concentric cylinder	273 to 973	Atmospheric pressures
5	Ikenberry, Lynn D., and Rice, Stuart A.	Steady state concentric cylinder	99 125.5 150.25 175.15 200.2 235.0	1.6 to 393 24 to 500 25.9 to 500 50.6 to 500 2.8 to 500 24.6 to 500
6	Golubev, I.F. (1963)	Steady state concentric cylinder	196.1 273.95 298.65 349.95 431.15 508.15	1 to 50 and 100 to 400 1 to 608 1 to 608 1 to 608 1 to 608 1 to 608

TABLE 1 (continued)

Reference No.	Author(s)	Method Adopted	Temperature Range in °k	Pressure Range in Bars
7	Carmichael, L.T. Reamer, H.H., and Sage, B.H. (1966)	Steady state concentric sphere	277.59	1 to 345
			310.93	1 to 345
			377.59	1 to 345
			410.93	1 to 345
8	Misic, D. and Thodos, G. (1966)	Steady state concentric cylinder	275.05	1 to 580
			297.75	1 to 575
			316.75	1 to 582
			348.25	1 to 585
9	Rosenbaum, B.M. and Thodos, G. (1967)	Steady state concentric cylinder	335.6	3.78 to 682
			371.3	4.47 to 697
			403.8	4.47 to 652
			435.3	4.47 to 626
10	Tufeu, R., Le Neindre, B. and Burg, P. (1967)	Steady state concentric cylinder	298.15	1 to 1162
			368.15	1 to 1023
			406.15	1 to 1147
			420.15	1.4 to 1017
			465.15	1.4 to 1025
			484.15	5.5 to 1128
			523.15	1.4 to 949
			571.15	1 to 1247
			575.15	1 to 1035
			621.15	1.4 to 1022
			648.15	5 to 1211
			725.15	10 to 890
11	Gonzales, M.H. Subramaniam, T.K., Kao, R.L. and Lee, A.L. (1968)	Steady state axial flow cylinder	122.8	5 to 100
			145.1	13 to 100
			172.8	27 to 100

TABLE 9 THERMAL CONDUCTIVITY VALUES FOR METHANE IN (watt/m°K)x 10^{3}

Temperature (°K)	PRESSURE (bars)															
	1	Psat	25	50	100	200	300	400	500	600	700	800	900	1000	1100	1200
100	11.56	205.57	206.95	208.83	214.07	230.42	254.63			1						
120	13.22	173.43	175.21	179.21	185.18	197.00	208.66									
140	15.03	144.69	147.48	151.17	158.21	170.97	181.97	193.10	202.0							
160	16.98	118.2	119.75	124.05	132.33	147.50	160.15	171.84	183.72	195.00						
180	19.07	92.84	–	96.51	105.28	123.95	137.86	149.13	159.87	172.80	188.24					
200	21.30		61.34	69.16	82.95	104.22	119.17	130.34	140.24	151.39	167.58					
220	23.66		45.57	53.76	68.15	90.13	105.23	116.10	125.38	135.71	149.73					
240	26.14		34.99	43.09	57.34	79.13	94.06	104.67	113.48	123.00	135.77	2				
260	28.74		33.40	38.50	50.05	69.20	85.40	95.77	104.22	112.99	124.30					
280	31.59		32.76	36.51	46.50	64.50	78.98	89.19	97.33	105.40	115.13					
300	34.49		35.01	38.24	45.62	60.75	74.54	84.44	92.49	101.10	108.50					
320	37.21		37.62	40.30	46.69	59.42	71.82	81.47	89.98	98.38	105.50					
340	39.92		40.27	42.67	48.20	59.45	70.00	79.68	88.48	96.42	103.55					
360	42.85		43.24	45.32	50.14	60.10	69.71	78.77	87.25	95.17	102.54					
380	46.30		46.41	48.22	52.45	61.34	70.12	78.61	86.76	94.58	102.06	109.21				
400	49.75		49.77	51.37	55.10	63.09	71.16	79.13	86.94	94.58	102.04	109.32	116.44			
420	52.96		53.30	54.72	58.06	65.31	72.77	80.25	87.71	95.12	102.46	109.73	116.93	124.08	3	
440	56.53		56.98	58.27	61.30	67.95	74.88	81.93	89.02	96.14	103.27	110.40	117.55	124.62	131.71	
460	60.34		60.79	61.98	64.78	70.95	77.43	84.07	90.80	97.60	104.44	111.31	118.21	125.14	132.18	134.55
480	64.27		64.71	65.83	68.46	74.25	80.36	86.62	92.98	99.42	105.91	112.45	119.02	125.82	132.25	136.97
500	68.30		68.73	69.80	72.30	77.81	83.59	89.50	95.50	101.56	107.66	113.79	119.95	126.13	132.32	138.53
520	72.41		72.82	73.86	76.28	81.56	87.06	92.65	98.29	103.96	109.63	115.32	121.00	126.66	132.30	136.05
540	76.57		76.98	78.00	80.36	85.46	90.71	96.00	101.29	106.56	111.80	117.01	122.19	127.30	133.44	137.50
560	80.78		81.17	82.18	84.50	89.44	94.47	99.47	104.42	109.30	114.11	118.85	123.51	128.10	132.92	137.67
580	85.01		85.39	86.39	88.66	93.47	98.28	103.01	107.63	112.14	116.53	120.81	124.96	129.04	133.00	136.68
600	89.24		89.61	90.59	92.82	97.47	102.07	106.53	110.85	115.01	119.02	122.88	126.61	130.19	133.64	136.80
620	93.46		93.82	94.77	96.93	101.40	105.77	109.98	114.01	117.86	121.53	125.04	128.39	131.56	134.68	137.27
640	97.64		97.99	98.91	100.97	105.20	109.33	113.28	117.04	120.62	124.08	127.26	130.31	133.24	136.08	138.67
660	101.77		102.12	102.97	104.89	108.83	112.67	116.36	119.89	123.26	126.47	129.53	132.48	135.23	137.67	140.38
680	105.88		106.18	106.94	108.66	112.21	115.73	119.16	122.48	125.70	128.81	131.83	134.79	137.57	140.31	142.86
700	109.74		110.15	110.80	112.24	115.31	118.45	121.60	124.75	127.89	131.02	134.14	137.23	140.31	143.37	146.41
720	113.64		114.02	114.51	115.61	118.07	120.75	123.62	126.64	129.78	133.05	136.45	139.91	143.48	147.14	150.88
740	117.30		117.76	118.06	118.72	120.42	122.59	125.15	128.07	131.31	134.86	138.69	142.79	147.12	151.70	156.51

TABLE 3. PRELIMINARY EXPERIMENTAL RESULTS

Pressure (bars)	Number of individual Measurements	Mean Temperature (°K)	Thermal Conductivity (watts/m°C)		Percentage deviation $\frac{(\lambda exp-\lambda cor)}{\lambda exp}$ x 100
			λ_{exp}	λ_{cor}	
578	4	291.31	106.72	96.85	+9.25
414	3	291.31	95.65	86.68	+9.59
313	3	291.31	84.32	76.21	+9.61
207	3	291.31	70.15	62.63	+10.7
553	3	258.52	120.46	109.43	+9.12
401	5	258.52	104.92	96.70	+7.82
311	4	258.52	94.75	87.23	+7.91
207	4	258.52	79.21	72.75	+8.21

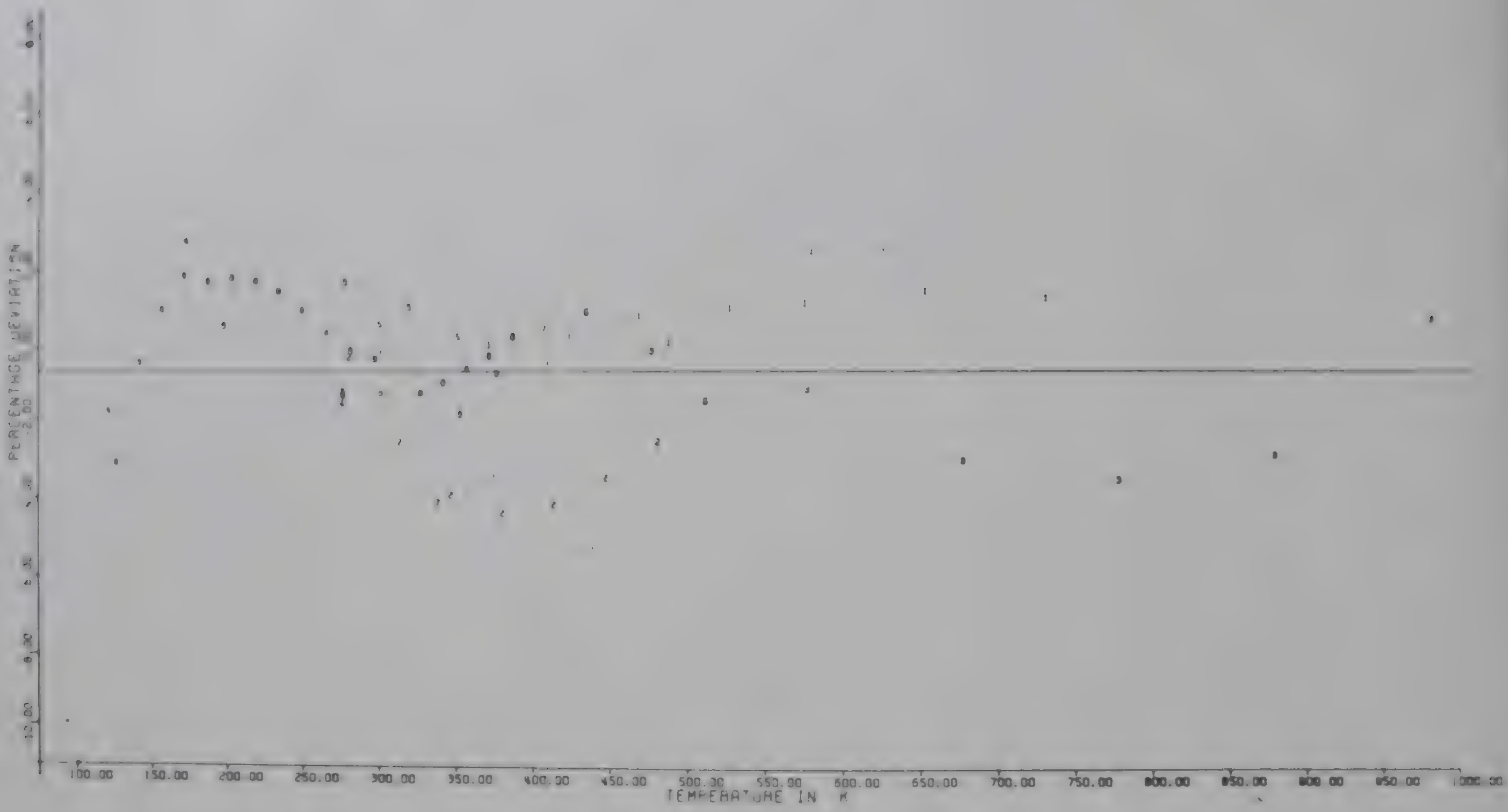


Figure 1. Deviation plot for atmospheric pressure data [Equation (2.1.1)]

Legend

1 – 298.15°K to 725.15°K [10]
2 – 277.59°K to 477.59°K [7]
3 – 273.15°K to 973.15°K [4]
4 – 120.45°K to 273.15°K [3]
5 – 275.05°K to 348.25°K [8]
6 – 273.95°K to 508.15°K [6]
7 – 335.55°K to 435.35°K [9]
8 – 96.76°K to 383.57°K [2]

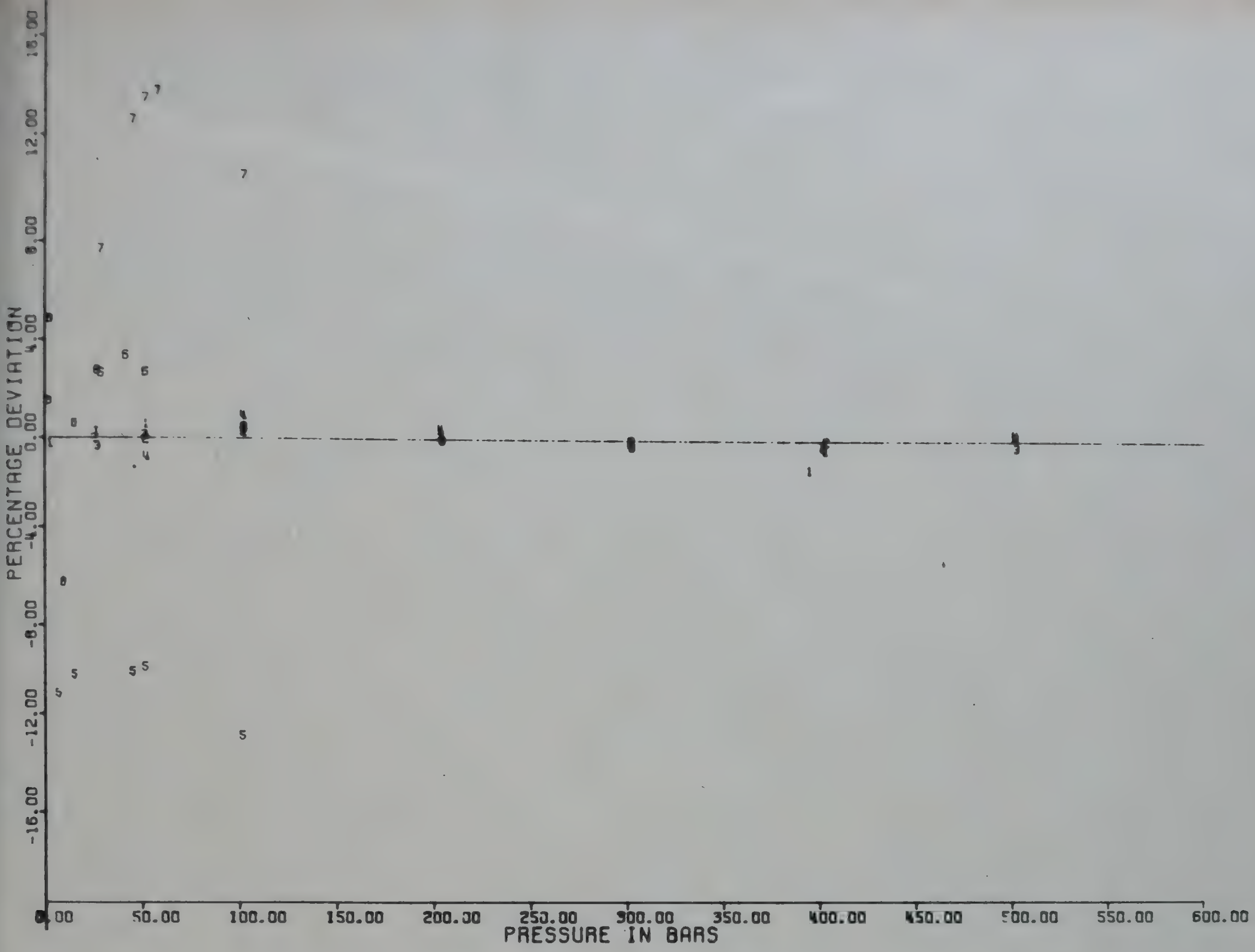


Figure 2. Deviation plot for Liquid State data.
[Equation (2.2.5)]

Legend

1 - 99.0°K [5]

2 - 125.5°K [5]

3 - 150.3°K [5]

4 - 175.2°K [5]

5 - 122.8°K [11]

6 - 145.1°K [11]

7 - 172.8°K [11]

8 - λ_{sat} values [1]

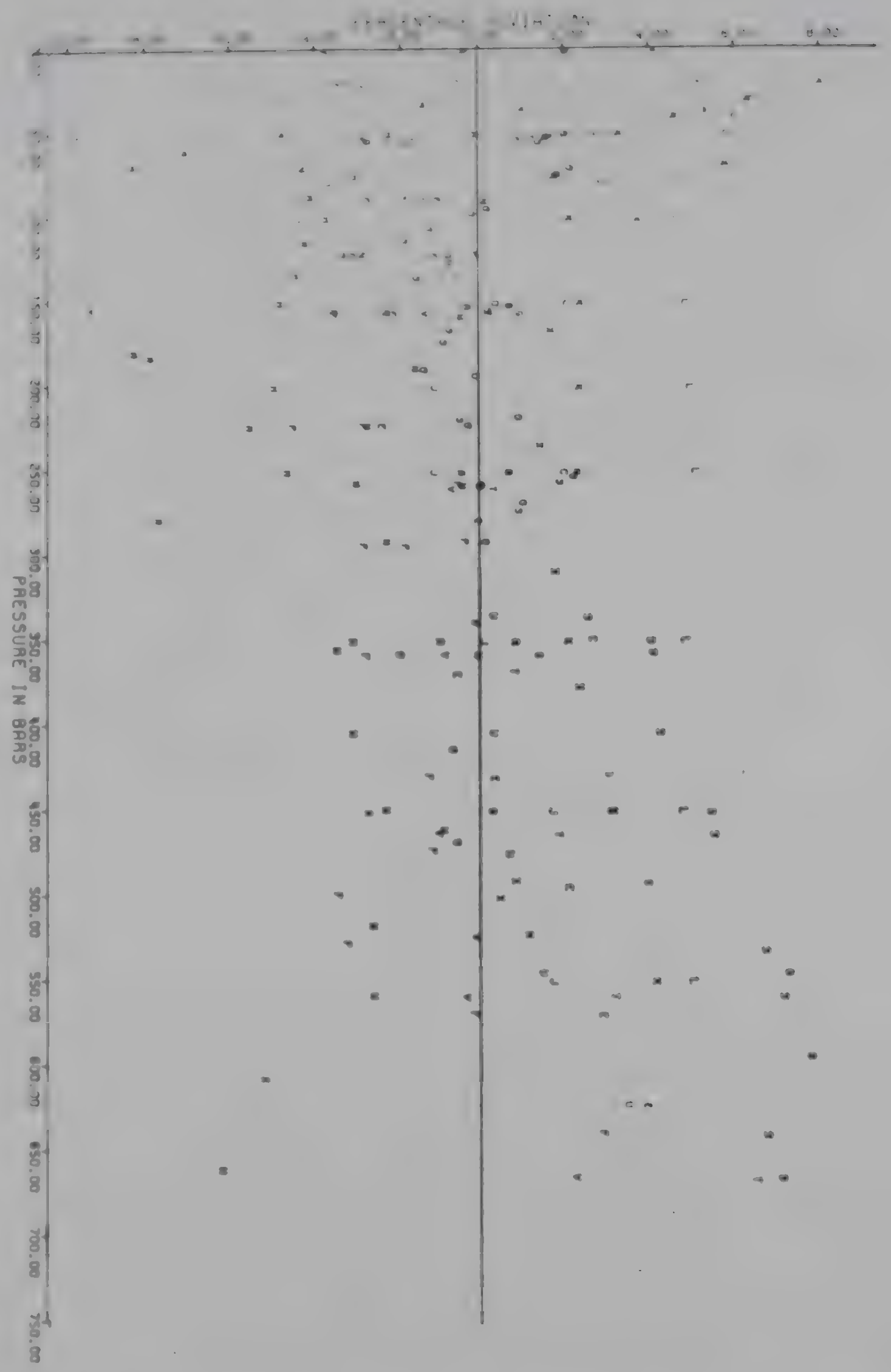


Figure 3. Deviation plot for Super Critical State data.
[Equation (2.3.1)]

Legend

A	–	150.25°K	[5]
B	–	175.15°K	[5]
C	–	220.2°K	[5]
D	–	235.0°K	[5]
E	–	335.6°K	[9]
F	–	371.25°K	[9]
G	–	403.8°K	[9]
H	–	196.15°K	[6]
J	–	273.95°K	[6]
K	–	298.65°K	[6]
L	–	349.95°K	[6]
M	–	275.05°K	[8]
N	–	297.75°K	[8]
O	–	316.75°K	[8]
P	–	348.25°K	[8]
S	–	298.15°K	[10]
T	–	368.15°K	[10]
U	–	406.15°K	[10]
V	–	420.15°K	[10]
W	–	465.15°K	[10]
X	–	484.15°K	[10]

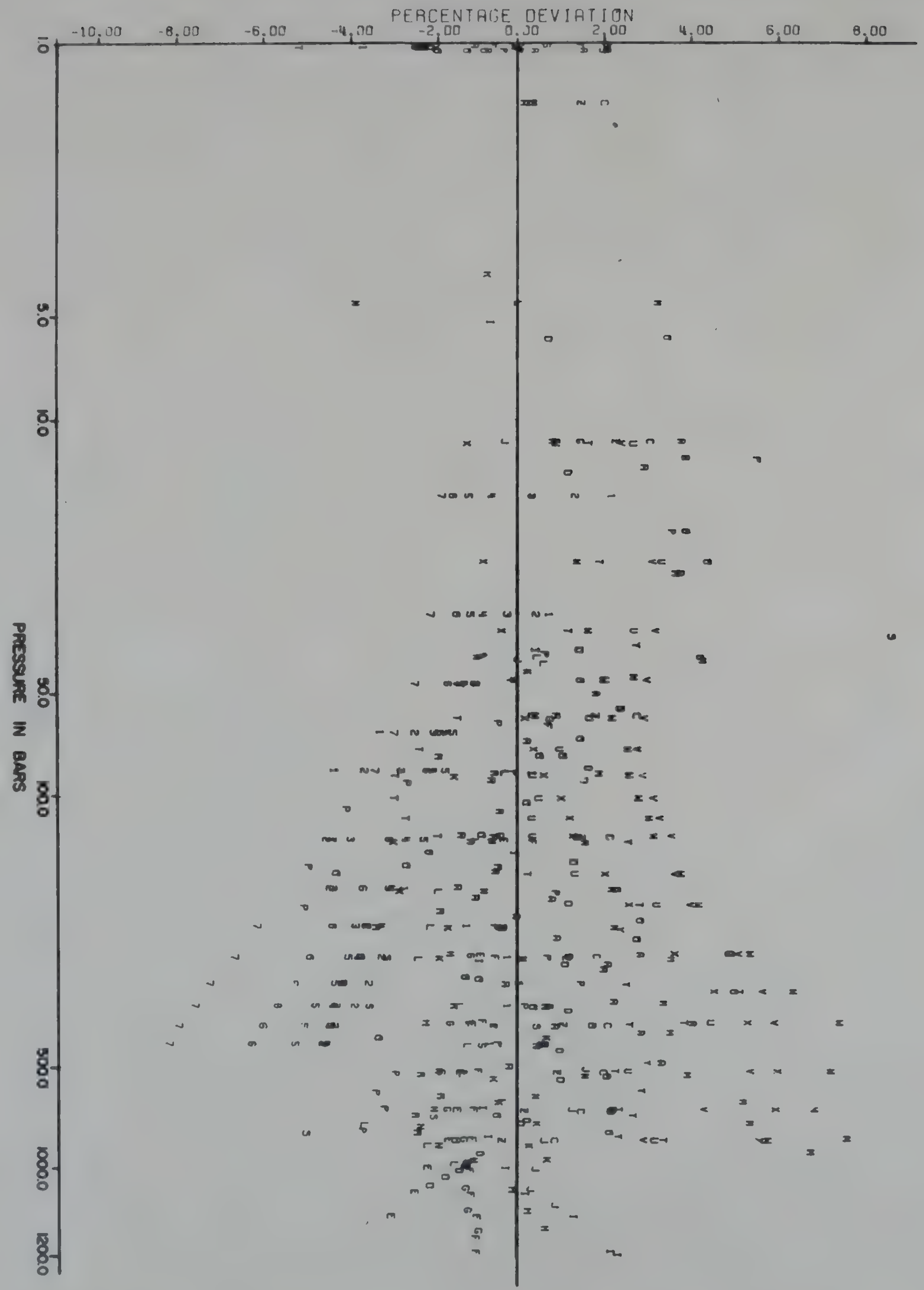


Figure 4. Deviation plot for Vapour State data. [Equation (2.4.1)]

Legend

A	-	298.15°K	[10]
B	-	368.15°K	[10]
C	-	420.15°K	[10]
D	-	484.15°K	[10]
E	-	523.15°K	[10]
F	-	571.15°K	[10]
G	-	575.15°K	[10]
H	-	621.15°K	[10]
I	-	648.15°K	[10]
J	-	725.15°K	[10]
K	-	335.55°K	[9]
L	-	371.25°K	[9]
M	-	403.75°K	[9]
N	-	435.35°K	[9]
O	-	275.05°K	[8]
P	-	297.75°K	[8]
R	-	316.75°K	[8]
S	-	348.25°K	[8]
T	-	273.95°K	[6]
U	-	298.65°K	[6]
V	-	349.95°K	[6]
W	-	431.15°K	[6]
X	-	508.15°K	[6]
Y	-	406.15°K	[10]
Z	-	465.15°K	[10]
1	-	277.59°K	[7]
2	-	310.93°K	[7]
3	-	344.25°K	[7]
4	-	377.59°K	[7]
5	-	410.93°K	[7]
6		444.26°K	[7]
7		477.59°K	[7]

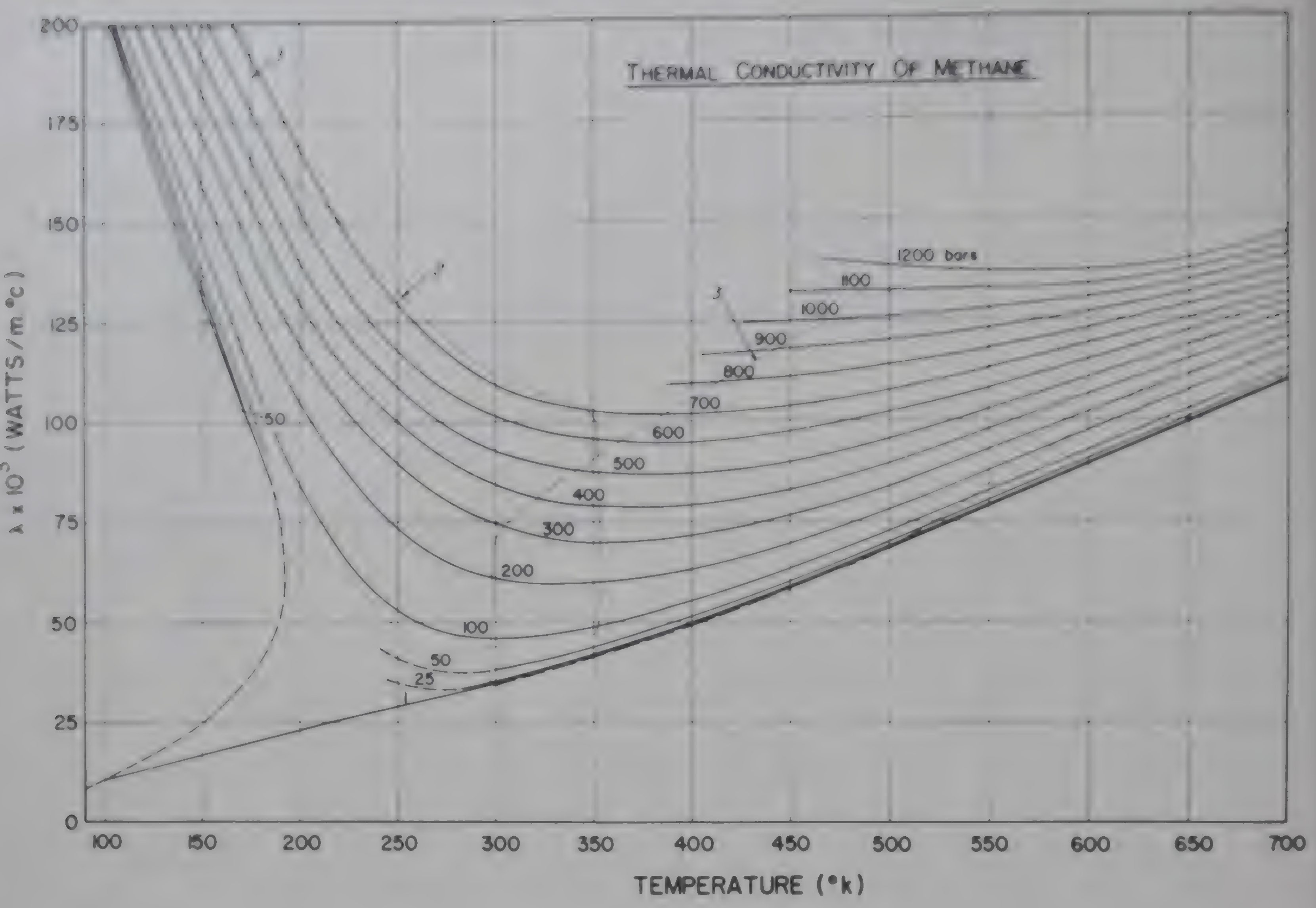


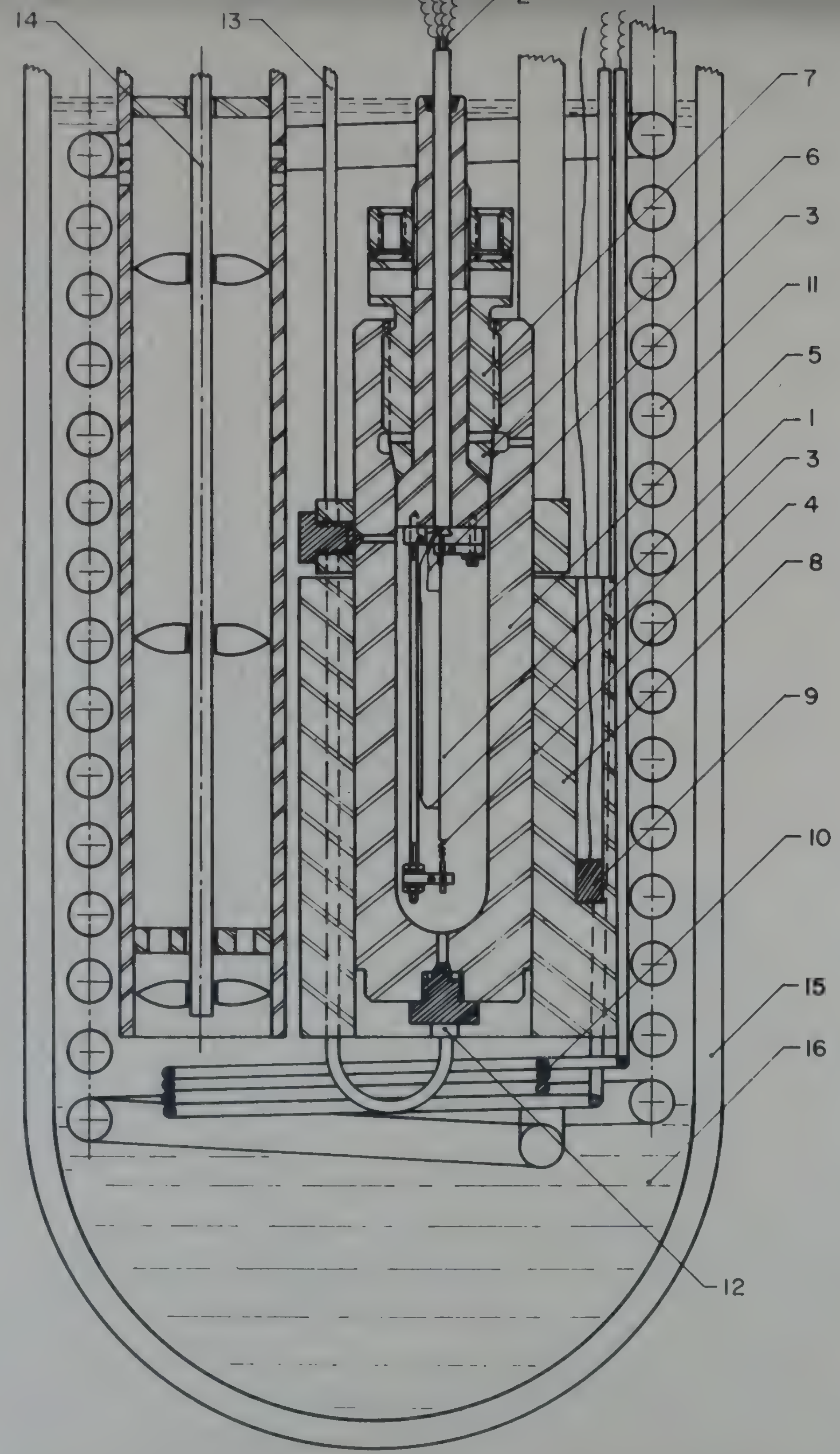
Figure 5. Thermal Conductivity of Methane - Smoothed values

Legend

1 - Liquid State [Equation (2.2.5)]

2 - Super Critical State [Equation (2.3.1)]

3 - Vapour State [Equation (2.4.1)]



Legend

1 - Line Source
2 - Electrical leads to measuring circuit
3 - Potential leads
4 - Tension springs
5 - Pressure vessel
6 - Seal ring
7 - Lock nut
8 - Aluminium Jacket
9 - Quartz Thermometer
10 - Heating coil
11 - Liquid Nitrogen cooling coil
12 - Pressure line
13 - Inlet for experimental gas
14 - Stirrer
15 - Dewar
16 - Cryostat Liquid - Di-methyl Butane

Figure 6. Cross-section of High Pressure Conductivity cell and Cryo-stat

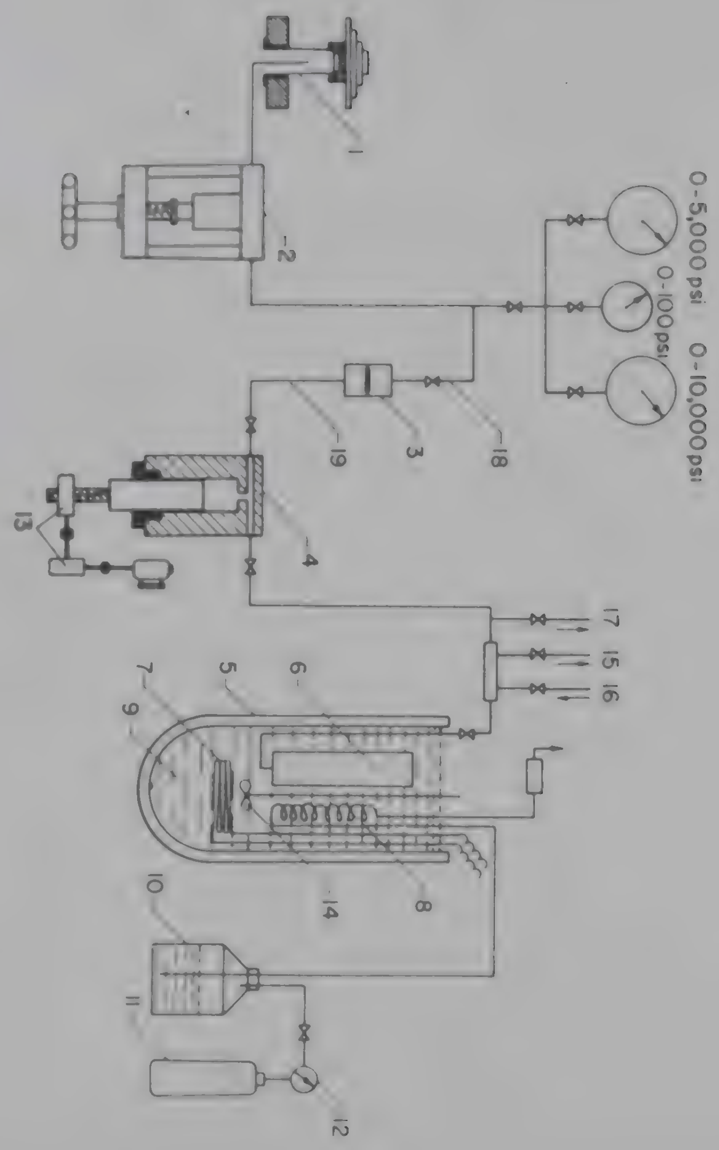

0-5,000 psi
0-100 psi
0-10,000 psi
1
2
3
4
5
6
7
8
9
10
11
12
13
14
15
16
17
18
19

Figure 7. High pressure System.

Legend

1	-	Standard Budenberg-dead weight tester
2	-	Screw press
3	-	Differential pressure null detector
4	-	Ruska - Positive displacement pump
5	-	Cryostat bath
6	-	High pressure conductivity cell
7	-	Heating coil
8	-	Liquid Nitrogen cooling coil
9	-	Cryostat Liquid - De-methyl Butane
10	-	Liquid Nitrogen dewar
11	-	Nitrogen cylinder
12	-	Nitrogen regulator
13	-	Gear boxes to drive Ruska pump
14	-	Stirrer
15	-	To Vacuum pump
16	-	For charging gas
17	-	Vent
18	-	Oil line
19	-	Gas line

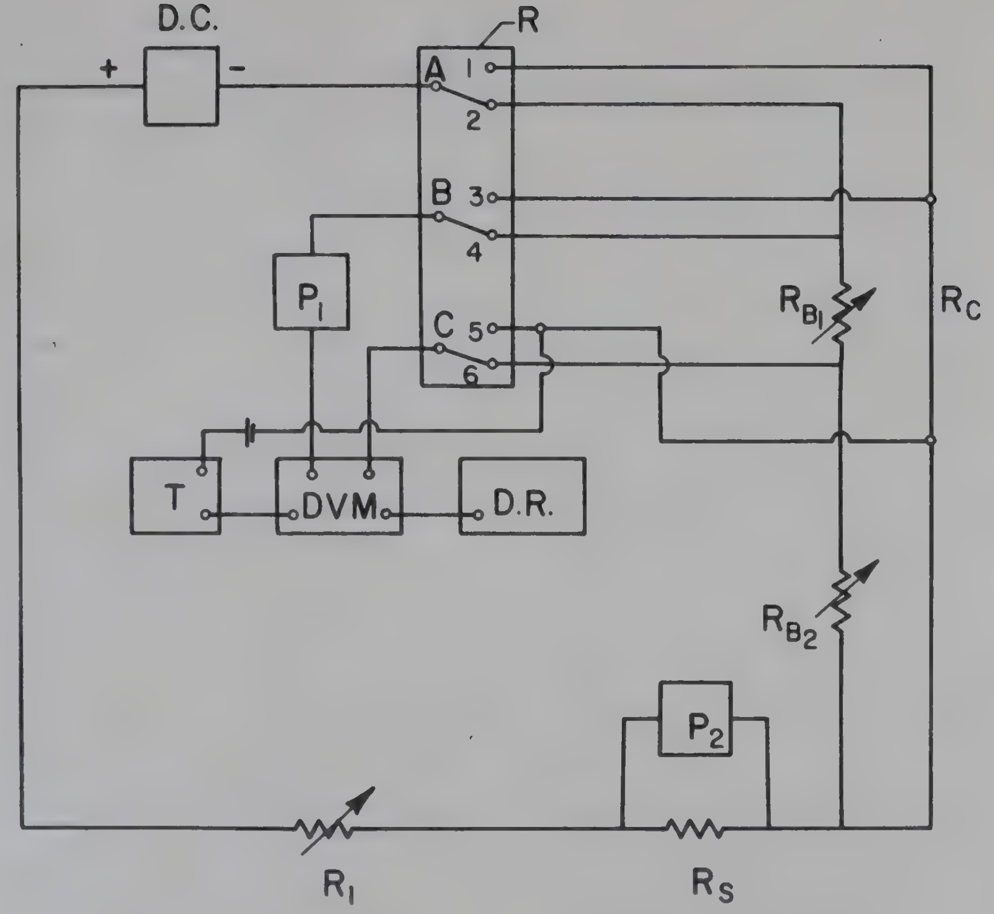

D.C.
+
-
R
A
1
2
B
3
4
C
5
6
P_1
T
DVM
D.R.
R_{B_1}
R_C
R_{B_2}
P_2
R_1
R_S

Figure 8. Schematic Electrical Circuit.

Legend

P_1, P_2 -	Potentiometers
DVM -	Integrating Digital Voltmeter (HP-2402A)
DR -	Digital Recorder (HP-5050B)
T -	Trigger (Set for 20 measurements per sec.)
R -	Fast response relay
R_c -	Cell resistance
R_{B1}, R_{B2} -	Ballast resistance
R_s -	Standard resistance
DC -	Direct Current power supply (HP-6112A)

PROBLEM OF HYDRATE FORMATION IN NATURAL GAS PROCESSING EQUIPMENT

By A. Lapin and S.J. Cinnamon*

ABSTRACT

This paper studies the problem of hydrate formation in natural gas in the temperature range of 0 to 21°C and at 42 kg/cm^2 pressure. Hydrate formation is of importance in the operation of natural gas liquefaction plants since the minimum temperature to which the natural gas may be cooled before requiring drying is determined by hydrate formation temperature. A test apparatus was built to simulate conditions in an LNG plant and to measure experimentally hydrate formation temperature as a function of flow rates and composition. Methods for calculating the temperature of hydrate formation were considered and a comparison was made between the calculated and experimentally measured values. The average difference between the two was less than 1.3°C. Observations were also made of the conditions that promoted hydrate formation and these are described in the paper in some detail.

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INTRODUCTION

In the early years of natural gas transmission "freezing" in the pipelines was assumed to be the result of the formation of ice, this assumption being made whether the ambient temperature surrounding the pipelines was below or above the freezing point of water. It was generally explained that the ice temperature of 0°C was achieved in the pipelines as a result of cooling by local expansion. The history of gas hydrates is covered in some detail by Deaton and Frost (8, 9)*. It was not until 1934 that Hammerschmidt (10) clarified the subject of hydrates through a series of tests performed at the newly installed gas cooling plant owned by the Texoma Natural Gas Company, Fritch, Texas. The work was initiated when plant equipment started to plug with ice, even though the flowing gas was originally cooled to a temperature of 4°C in order to reduce the gas dew point sufficiently to prevent condensation of water in the line at the operating conditions. In this instance the formation of ice occurred even though the gas temperature had been maintained above 4°C. It was also established that there was not enough local cooling by throttling or other causes to produce 0°C temperatures anywhere in the line. The fact that freeze-ups could occur in pipelines well above 0°C was definitely established and was of great concern to the natural gas industry. A relatively large research effort was initiated about that time to provide a basic understanding of gas hydrates as well as engineering solutions for the prevention of freeze-up in lines and to allow reliable pipeline operation. The hydrate formation problem is also of great interest to processing equipment designers and operators. Determination of hydrate formation temperature at the process pressure conditions will establish the process drying requirements and thus affect the economics of the operation.

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*Numbers in parenthesis refer to "Bibliography"

GAS HYDRATES

The gas hydrates as usually obtained in pipelines or experimental equipment, look like packed snow and have a density dependent on the pressure and temperature of formation. Under certain conditions the hydrates look more like ice crystals than packed snow. Early studies of these solids showed that they were compounds of water and hydrocarbons.

Hammerschmidt (10) experimentally studied the formation of hydrates with pipeline natural gas treated to remove the excess water and oil vapors. He determined conditions at which hydrates formed in an experimental unit consisting of a pyrex glass tube in which the treated gas flowed under pressure. He also obtained the temperature at which decomposition occurred in a non-flow condition. Conditions favoring hydrate formation were identified as a result of this effort and work by Katz (12, 13), Ceccotti (7) and others (3, 5, 6, 8, 9, 15, 16, 19, 20, 22). It was determined that gas hydrate formation depends primarily upon the temperature, pressure and composition of the gas. Experimentation was performed to establish the relationship between these three factors by recording conditions that favored either the formation or the decomposition of the gas hydrates. During this work it was shown that gas hydrates would not always form when the primary factors, namely temperature, pressure, and composition, were in the right relationship to favor their formation. It was established that secondary causes similar to those promoting crystallization, such as seeding, subcooling, and pressure disturbances, were required to insure hydrate formation.

In view of the fact that hydrates exist only in a narrow range of temperature and pressure, it is extremely difficult to properly sample them and thus determine their chemical compositions. It was nevertheless established that the gas hydrate is not a chemical compound, but is a clathrate or an inclusion compound. Brown (4) describes inclusion compounds as "chemical combinations in which one component fits into a cavity in the other. When the size and shape of the cavity in the first molecule just match the form of the 'guest', a combination of appreciable strength can occur. No ordinary chemical bonds between the atoms of the 'host' and 'guest' are needed." Marshall (16) defines the gas hydrate as "the stable crystalline ice-like complex formed from the lower molecular weight non-polar and slightly polar compounds and water when a mixture of these is subjected to external pressure". In the case of gas hydrates two different structures exist, I and II, each having two different size cavities. Sizes of solute molecules which fit each one of the two cavities are tabulated in the "Technical Data Book - Petroleum Refining" (1). A review of the hydrate structure as determined by x-ray defraction techniques is made by Saito et al (20) together with a theoretical development of the subject. Saito concurs that the gas hydrates crystallize in either of two cubic structures, I and II, and reports that the unit cell of structure I contains 46 water molecules enclosing two types of cavities: 6 pentagonal dodecahedral cavities and 2 tetrakaidecahedral cavities having two opposite hexagonal faces and 12 pentagonal faces serving to connect the regular dodecahedra. The unit cell of structure II is composed of 136 water molecules arranged to form 16 smaller cavities and 8 larger ones. Most of the work done to establish the structure of the hydrates and their composition was performed with a limited number of hydrocarbon-water systems. In most cases, this work was performed in pressure cells in which pressure and temperatures were adjusted to cause hydrate formation. Hydrates so formed were then analyzed and their composition determined. Hydrate compositions were also calculated from equilibrium data with some degree of success. However, because of the experimental difficulties, discrepancies exist between values reported by various investigators.

Since the main difficulty resulting from hydrate formation in pipelines or process equipment is plugging, various attempts were made to inhibit the formation of hydrates by different techniques. The most obvious one is the removal of water from the natural gas thus preventing the formation of hydrates. Several methods were considered for this purpose such as refrigeration of the gas and removal of the condensed water. An example of this is described in Henry's patent (11). Other conventional refrigeration systems which lower the dew point of the gas to the required level were also studied. Methods for the inhibition of hydrate formation by the addition of liquid hydrocarbons and glycols were considered. Scauzillo (21) describes some laboratory work in which this approach was taken. He concludes that the presence of liquid hydrocarbons in a natural gas lowers the hydrate formation temperature of the gas and that glycol solutions will also inhibit hydrate formation. Additional information on inhibition of hydrate formation is given in the literature (1, 18).

As a result of the study of gas hydrates, several applications taking advantage of their composition and structure were suggested. One interesting application was advanced by Barduhn, et al (2) as the feasibility of demineralizing sea water by hydrate formation and decomposition processes. Another idea was presented by Miller and Strom (17) regarding the possibility of storing natural gas in the form of solid hydrate to take advantage of the increased density of the gas in the hydrate form and thus have a more efficient storage system.

PREDICTION OF CONDITIONS AT WHICH HYDRATE FORMATION OCCURS

A major problem confronting pipeline and process equipment designers and operators is the ability to predict conditions at which hydrates will form in either pipelines or process lines and equipment. Accurate prediction of these conditions will enable appropriate action to prevent plugging of the lines and valves in the systems involved. The majority of the work done to enable the prediction of these conditions was performed by D. L. Katz and associates (6, 12, 13, 15, 22) in the nineteen forties. Through experimental work and theoretical treatment of the data, Katz et al established a series of K-value charts and methods of calculations enabling the prediction of conditions at which hydrate formation will occur. The results of this work were widely published in the Natural Gas Industry (1, 14, 18). However, the correlations developed by Katz were based on experimental data mostly obtained in non-flow glass or glass-windowed equipment with simple water-rich systems. Plant operation, however, involves continuous flow of complex mixtures at relatively high mass velocities.

EXPERIMENTAL PROGRAM

This experimental program was based on conditions simulating typical process conditions in a natural gas liquefaction plant, with the main objective of providing new data to be used in the verification of the validity of the published methods for the prediction of hydrate formation temperatures.

Experimental Loop and Equipment

The experimental flowsheet is shown on Figure 1. A brief description of the experimental loop is as follows.

The heart of the test loop was cooling coil #2 which was made of 4.77 mm O.D. x 0.81 mm wall copper tubing and was immersed in a cold water bath. It consisted of four 30.5 cm sections. The coil had crosses every 30.5 cm for instrumentation purposes. At each cross there was an internal and an external thermocouple probe and an internal pressure probe. The internal probes were in the gas stream and the external probes were soldered to the tube wall. An additional run was made using a 1.22 m long smooth continuous coil of the same diameter to establish whether or not the pressure taps and thermocouples in the coil had any effect on hydrate formation.

Heating coil #1 consisted of a 3 m length of 6.35 mm O.D. x 1.24 mm wall copper tubing.

The saturator and separator were both 2.86 cm O.D. x 1.65 mm wall copper tube, 30.5 cm long. The saturator was filled with 6.35 mm alumina balls.

The instrumentation consisted of:

Pressure gauges: P1, P2, P3, P4, P5, P6
Bath thermocouples: T11, T12, T13
Wall thermocouples: T2, T4, T6, T8, T10
Fluid thermocouples: T1, T3, T5, T7, T9
Flowmeter: 5 SCFM Rotameter
Sample Taps: S1, S2

Experimental Technique

The following three different test gases were investigated.

Component	Feed Composition, % CASE I	CASE II	CASE III
N_2	1.70	1.58	1.47
CH_4	72.78	67.69	62.49
C_2H_6	14.50	13.50	12.47
C_3H_8	7.63	14.10	20.62
C_4H_{10}	2.65	2.45	2.26
C_5H_{12}	0.63	0.58	0.58
C_6H_{14}	0.11	0.10	0.11
	100.00	100.00	100.00

All tests were run at 42.9 kg/cm^2 abs pressure with the gases saturated with water at 37.8°C. Most of the runs were performed with a mass velocity through the cooling coil of 591,000 kg/hr m^2 which was limited by the Corblin compressor capacity. Several runs were performed at reduced mass velocities.

The test gas was compressed to 42.9 kg/cm^2 abs, heated to 37.8°C and saturated with water. The gas passed from the saturator to the cooling coil which was immersed in a water bath. Two experimental techniques were used to form hydrates. The first one involved slowly lowering the cooling bath temperature until hydrates started to form. This was indicated by an increase in pressure drop across the cooling coil, and by visually observing hydrates in the Jerguson Sight Glass. Because of the long time required for this procedure, an alternate procedure was tried. The cooling coil was removed from the bath and ice was rubbed on the side of the coil for a few seconds to locally induce hydrate formation. The cooling coil was then returned to the water bath. If hydrates persisted and continued to form, the bath temperature was below the hydrate formation temperature; and if they disappeared, the bath was above the hydrate formation temperature. The coil was removed from the bath and the bath temperature increased or decreased as required. Ice was again rubbed on the coil until hydrate formation began and the coil was immersed in the bath. This procedure was repeated until the hydrate temperature was determined. This technique required less time than the initial one and gave the same results.

The gas flow through the cooling coil was varied by changing the position on the bypass valve V2, to establish if the gas flow had any effect on the hydrate formation temperature.

Observations

The hydrates which collected in the Jerguson Sight Glass resembled ordinary wet snow in appearance. Hydrates did not form until the test gas had been subcooled by at least 2°C. However, once hydrate formation started the lines readily plugged up. At this point the cooling water was turned off. The bath was slowly allowed to warm up until all hydrates disappeared. It was assumed that the temperature at this point would be the hydrate formation temperature. Considerable time was required for the hydrates to thaw out. The flow frequently became pulsating and there was considerable pressure drop across the cooling coil. After the system had thawed out, the cooling water was again turned on and run until hydrate formation was indicated. This time it was not necessary to subcool the gas to form hydrates. It is assumed that all the hydrate crystals did not melt on the thawing cycle and provided sites for hydrate formation.

Results and Discussion

The following hydrate formation temperatures were obtained at a 42.9 kg/cm^2 abs pressure

Case I	15.6°C
Case II	16.7°C
Case III	16.7°C

These experimental hydrate formation temperatures were identical to the hydrate decomposition temperatures obtained in some of the runs.

It was more difficult to initially form hydrates in the set of tests which used a single 1.22 m long smooth coil. However, the experimentally measured hydrate formation temperature was the same as for the coil consisting of four 30.5 cm sections.

The hydrate formation temperature was found to be independent of the gas flow rate.

All the thermocouples that were used to measure the hydrate formation temperatures were carefully checked to insure accurate readings. It is felt that the measured temperatures for hydrate formation are good to within ± 0.3°C. Pressures recorded were accurate to within ± 0.7 kg/cm^2 at 42 kg/cm^2.

COMPARISON BETWEEN CALCULATED AND MEASURED HYDRATE FORMATION TEMPERATURES

Three methods for the calculation of hydrate formation temperatures were used:

Method I - Vapor-Solid Equilibrium Calculations
(Figs. 5.42, 5.43, 5.44, 5.45, Ref. 14 or Figs. 7, 8, 9, 10 Ref. 18)

This method is analogous to the dewpoint calculation for a multicomponent system using liquid-vapor equilibrium constants.

It has been postulated that hydrate formation conditions could be calculated from empirically determined vapor-solid equilibrium constants.

$$K_{v-s} = \frac{y}{x_s}$$

where y = mole fraction of hydrocarbon in the gas on a water-free basis

x_s = mole fraction of hydrocarbon in the solid on a water-free basis

These constants were determined by Carson and Katz (6) for methane, ethane, propane, and isobutane, and presented in various handbooks (14, 8). In the presence of lighter hydrocarbons, the equilibrium constants for n-butane, for low concentrations of n-butane, may be taken as those of ethane. For nitrogen and hydrocarbons heavier than butane, the equilibrium constants are taken as infinity, since they form mixed hydrates with difficulty. The equilibrium constants are assumed to be functions of pressure and temperature only, but actually they should show variation with hydrocarbon composition. The conditions for initial formation are obtained by satisfying the relationship:

$$\Sigma \frac{y}{K_{v-s}} = 1.0$$

a procedure analogous to the dewpoint calculation for complex gaseous mixtures.

Method II - Gas Gravity
(Fig. 5.40, Ref. 14 or Fig. 1, Ref. 18)

In this procedure the gas gravity (air = 1.0) has been utilized as a correlating parameter. The calculated hydrate temperature depends on the specific gravity and pressure of the gas.

Method III - Gas Gravity with N_2 Correction
(Fig. 5.41, Ref. 14)

Method III is similar to Method II. It includes the effect of small concentrations of nitrogen on hydrate formation conditions of natural gases.

Accurate calculation of the hydrate temperature requires some knowledge about the hydrate temperature dependence on the vapor composition. In Method I the K-values are assumed to be independent of composition and in Method II the composition dependence is incorporated into the specific gravity of the vapor. Therefore, neither is exactly correct.

The following table gives the measured hydrate temperatures and the corresponding calculated hydrate temperatures. Three independent hydrate formation calculations were made for each case.

Hydrate Formation Temperatures
°C @ Pressure = $42.9\ kg/cm^2$ abs.

	Experimental Measurement	Calculated Temperatures		
		Method I	Method II	Method III
Case I	15.6	17.1	15.6	17.6
Case II	16.7	17.8	16.1	17.8
Case III	16.7	18.1	16.4	18.1

Since the agreement between the experimental results and the calculated hydrate formation temperature is close, it is felt that the correlating parameters can be used to predict the hydrate formation temperature to within ± 1.5°C. It appears that Methods I and III predict about the same slightly high values for hydrate formation temperature while Method II predicts closer values to the experimental measurements, but slightly lower.

Detailed liquid vapor splits for the three test gases as calculated from K-values are tabulated below:

Vapor-Liquid Equilibrium

CASE I
Pressure - 42.9 kg/cm^2 abs
Temperature - 15.6°C

Component	Feed	Liquid	Vapor	K-Value
N_2	1.700	0.001	1.699	19.28632
C_1	72.780	0.109	72.671	4.75802
C_2	14.500	0.125	14.375	0.81680
C_3	7.630	0.176	7.454	0.30207
C_4	2.650	0.164	2.486	0.10832
C_5	0.630	0.098	0.532	0.03844
C_6	0.110	0.035	0.075	0.01517
Total, Mol	100.000	0.708	99.292	

CASE II
Pressure - 42.9 kg/cm^2 abs
Temperature - 16.7°C

Component	Feed	Liquid	Vapor	K-Value
N_2	1.580	0.003	1.577	15.82827
C_1	67.690	0.462	67.228	4.37744
C_2	13.500	0.465	13.035	0.84190
C_3	14.100	1.207	12.893	0.32110
C_4	2.450	0.486	1.964	0.12133
C_5	0.580	0.233	0.347	0.04490
C_6	0.100	0.062	0.038	0.01816
Total, Mol	100.000	2.918	97.082	

CASE III
Pressure - 42.9 kg/cm^2 abs
Temperature - 16.7°C

Component	Feed	Liquid	Vapor	K-Value
N_2	1.470	0.010	1.460	13.62311
C_1	62.490	1.434	61.056	4.13996
C_2	12.470	1.276	11.194	0.85234
C_3	20.620	4.689	15.931	0.33022
C_4	2.260	0.971	1.289	0.12903
C_5	0.580	0.387	0.193	0.04854
C_6	0.110	0.091	0.019	0.01991
Total, Mol	100.000	8.858	91.142	

Sample Calculation of Hydrate Formation Temperature

Case I - Vapor Composition Taken @ 15.6°C, 42.9 kg/cm^2 abs
Measured Temperature - 15.6°C

METHOD I

Component	Composition	$K_{62°F}$	M/K	$K_{63°F}$	M/K
N_2	1.699	∞	---	∞	---
CH_4	72.671	1.55	46.884	1.57	46.287
C_2H_6	14.375	1.10	13.068	1.21	11.880
C_3H_8	7.454	0.260	28.669	0.31	24.045
C_4H_{10}	2.486	0.148	16.797	0.17	14.624
C_5H_{12}	0.532	∞	---	∞	---
C_6H_{14}	0.075	∞	---	∞	---
Total, Mol	99.292		105.418		96.836
ΣM			99.292		99.292
△			6.126		-2.456

$$T = 62.0 + \frac{6.126}{6.126 + 2.456} = 62.7°F = 17.1°C$$

M.W. = 21.79

SpG = 21.79/28.96 = 0.752

	METHOD II	METHOD III
Hydrate Temperature -	15.6°C	17.6°C

CONCLUSIONS

1. All three methods of calculation considered can be used to predict the hydrate formation temperature within 1.5°C. The averaged differences between the calculated temperatures and the experimentally measured values were +1.3, -0.3, and +1.5°F for Methods I, II, and III respectively.

2. It is not difficult to form hydrates at the predicted hydrate temperature provided that some seed crystals are present or that some seeding is induced by subcooling (approximately 2°C) the gas for a short period of time.

3. Hydrate formation represents a serious problem. When plugging takes place it is usually severe. The assumption that a slight pressure buildup will force the hydrates through the tubes may not be valid. In the experimental work, pressure drops up to 42.2 kg/cm^2 across the hydrates were not uncommon.

4. The temperature at which hydrates form does not depend on the gas flow rate within the range of flows of this experiment.

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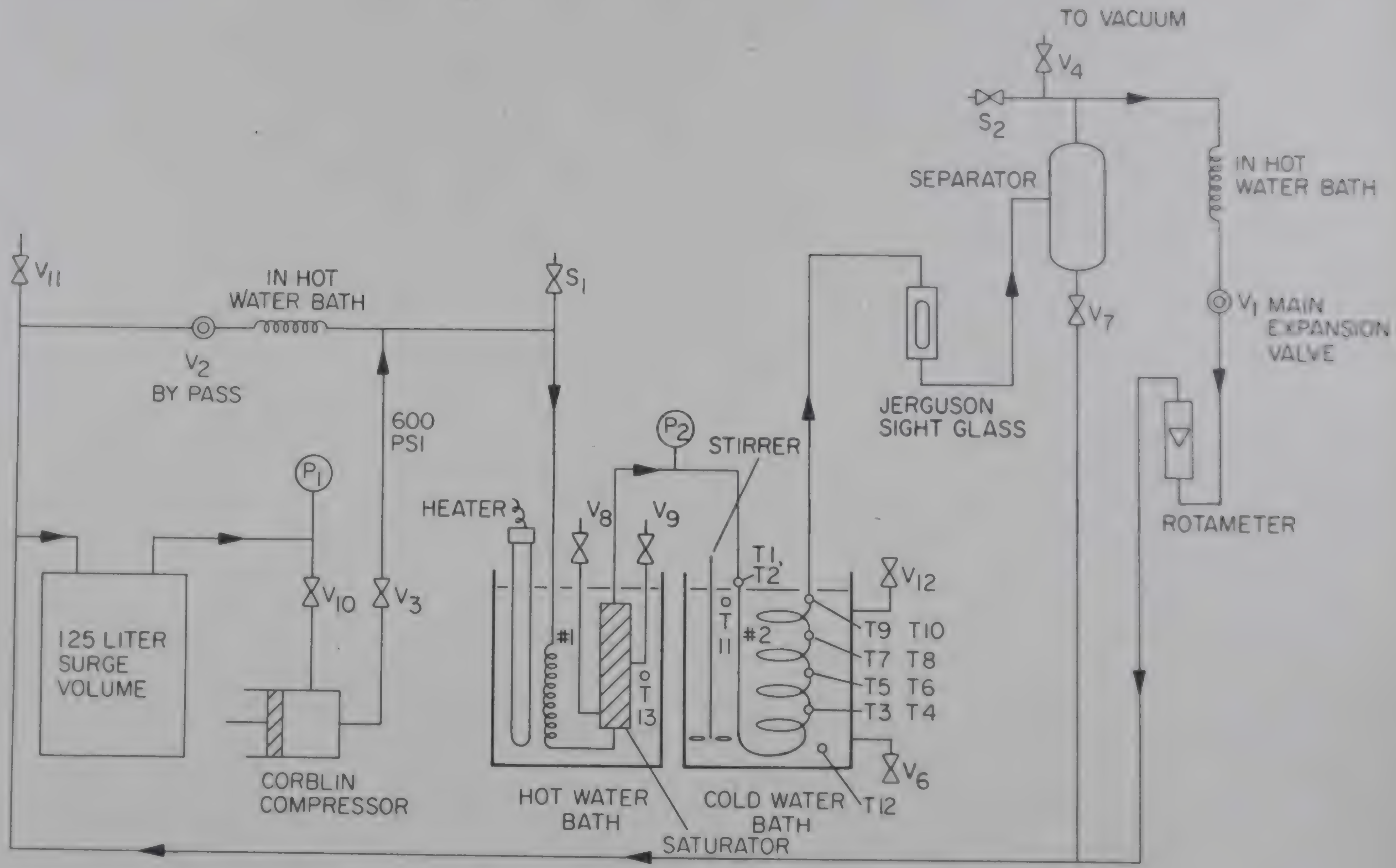
TO VACUUM
V_4
S_2
SEPARATOR
IN HOT WATER BATH
V_1 MAIN EXPANSION VALVE
V_7
ROTAMETER
JERGUSON SIGHT GLASS
V_{11}
IN HOT WATER BATH
V_2 BY PASS
S_1
600 PSI
P_1
P_2
STIRRER
HEATER
V_8
V_9
T1, T2
V_{12}
T9 T10
T7 T8
T5 T6
T3 T4
V_6
T12
#1
#2
T 11
T 13
125 LITER SURGE VOLUME
V_{10}
V_3
CORBLIN COMPRESSOR
HOT WATER BATH
SATURATOR
COLD WATER BATH

FIGURE I HYDRATE FORMATION TEST LOOP

A SECOND GENERATION LNG PEAK SHAVING CYCLE

By J. A. Snyder*

A number of liquefied natural gas peak shaving plants are in various stages of construction or operation in the USA. For the most part, the cycles utilized are either simple pressure letdown or fairly complex classic cascade cycles. Accumulated operating experience indicated a need for improvement in a number of areas, the most pronounced of which were odorant disposal, refrigerant storage and makeup, tank "weathering" or calorific adjustment problems, and the disposal of spent cleanup system regeneration gas.

With these problems in mind, we established the following ground rules for our response to the first user who was planning to purchase his second plant.

- --Centrifugal machinery is to be used for main drives and wherever else it is practical.
- --No gas is to be returned to the mains.
- --Gas composition in storage must be maintained at a constant composition.
- --Odorant disposal problems must be eliminated.
- --Cold blowers must be eliminated.
- --Calorific adjustment problems of return gas must be eliminated.
- --Operation must be simple.
- --Power consumption and capital investment considerations must be optimized.
- --Major machinery pieces shall not have speed increasing or reduction gears.

The feed stock available had the following design analysis, with swings of any constituent within the limits stipulated.

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*Airco/BOC Cryogenic Plants Corporation, New Jersey

Feed Gas Specifications

Constituent	Design Mol %	Maximum	Minimum
Helium	0.05	.07	.02
Nitrogen	3.00	3.94	1.92
Methane	88.85	90.00	83.80
Ethane	6.00	10.15	6.00
Propane	1.00	2.55	1.00
Iso-Butane	.05	.16	0
N-Butane	.05	.26	.02
Carbon Dioxide	1.00	1.00	.06
Water Content (#/MMcf)	13	40	2
Total Sulphur Content (gr/ccf)	2	5	.06
Hydrogen Sulphide (gr/ccf)	.1	.25	0
HHV	1051		
Inlet gas pressure	15 Atm.	28 Atm.	8 Atm.
Inlet gas temperature	30° C	32° C	10° C

The plant is required to liquefy 200,000 standard cubic meters of the gas per day. It is further stipulated that up to 38,000 liters per day of 99 percent pure liquid methane could be withdrawn from the plant.

One of our parent corporations, Air Reduction Corp. of New York, made the features of U.S. Patent #3,358,460 available to us.

This patent provides a very efficient cryogenic refrigeration or liquefaction cycle and yet allows the very desirable use of centrifugal compression and expansion equipment, and the use of plate fin heat exchangers.

The major inefficiency in a cryogenic refrigeration cycle can best be, perhaps over simply, described as heat exchanger warm end "delta T". It is simply desirable to have the refrigerant gas give up as much of its refrigeration as possible to the incoming process stream.

In the high pressure old fashioned plants requiring wound tube heat exchangers, this was not a complicated feat. Figure 1 illustrates parallel and straight warming and cooling curves for the two gas streams, but note that these gases are at 180 atm., a pressure not attainable with brazed aluminum exchangers.

As we reduce the pressures to practical working limits for commercial exchangers, the perverse nature of the stream being cooled is illustrated in Figure 2. The shape of the cooling curve, which is a plot of enthalpy change vs. temperature, is not one of a constant slope, and hence we discover the infamous "pinch effect" which has been the undoing of many a novice process designer. To provide sufficient temperature gradient to allow heat flow in the desired direction at the pinch points, it is necessary to open up the exchanger end point delta Ts, thus introducing undesirable losses.

The solution is to bend the shape of the heating curve to approximate that of the cooling curve, and this is the central feature of the reheat expansion cycle patent. By providing multiple expanders, and warming or reheating the gas between expanders, the composite heating curve is bent as is illustrated in Figure 3, thus significantly improving overall exchanger efficiency.

Having built and operated liquefiers using this principle for nitrogen circuits, we were thoroughly familiar with their overall characteristics. Examination of the feed gas analysis revealed a nitrogen content which would cause product and storage problems if the nitrogen were not removed. Thorough evaluation of the use of nitrogen as a refrigerant yielded the following advantages.

--The refrigerator side of the heat exchanger is not subject to change with a change in gas composition.

--Centrifugal machinery can be designed for a gas of known composition and not subject to variation. The thermodynamic qualities of nitrogen are well known throughout their entire working range. The centrifugal expanders will not operate in or near the wet region, thus eliminating mechanical difficulties experienced with natural gas expanders. With a clean gas, machinery performance is not likely to deteriorate with time.

--Nitrogen for use as a working fluid is taken from the gas feed stock, thus improving product quality. Compressor seal leakage is in effect a waste gas rather than expensive natural gas.

--With the lower temperatures available in the nitrogen cycle, storage tank boil-off is recondensed and reinjected into the storage tank, thus eliminating change in tank composition due to loss of the lower boiling constituents.

- --With the lower process temperature available, there is no flash loss when the LNG is injected into the storage tank. This is particularly important in maintaining tank composition since flash gas is primarily nitrogen and methane and tends to result in an accumulation of heavies in the tank.

- --With the lower temperatures available, the 99 percent pure methane which is delivered to transport vehicles can be subcooled, thus eliminating flash losses in the transport vehicle.

- --Nitrogen will be used to purge the coldbox or other appropriate areas. We believe this to be a very important safety consideration.

- --Nitrogen can be available for purging the insulation of the main storage tank.

- --Liquid nitrogen can be withdrawn from the plant for use elsewhere.

- --Greater cycle efficiency can be achieved because of better performance of machinery. Higher molecular weight of the refrigerant results in fewer stages of compression and expansion.

Thus, the cycle described in Figure 4 evolved.

A five bed molecular sieve process gas cleanup system was selected, after evaluation of a variety of water, carbon dioxide and odorant removal schemes. The drier beds are kept separate from the CO_2 removal beds for isolation purposes during shutdown. This prevents displacement of CO_2 by water in the unregenerated bed, and subsequent operating problems. With a three bed CO_2 removal system, one bed is in active use, one is being regenerated, and one is being cooled down at any given moment of time.

A bleed stream of purified gas is used for regeneration and heat for regeneration is provided by exchange against the gas turbine prime mover exhaust gas. A novel recirculation system for the regeneration gas increases its residence time and thus eliminates sharp peaks of odorant concentration in the effluent regeneration gas. This spent regeneration gas ultimately becomes fuel for the gas turbine driver. Purified process gas then is compressed, if necessary, and delivered to the cold box for liquefaction.

Before describing the liquefaction system, let us consider the refrigeration system.

As mentioned earlier, the refrigeration side of the liquefier is kept entirely separate and distinct from the gas processing side. The refrigerant gas selected is pure nitrogen which is derived from the natural gas feed stock itself. The closed loop nitrogen refrigerant system has the advantage of allowing the centrifugal expanders to operate in a completely dry region while still supplying subcooling for the LNG which is being sent to storage. Basically the refrigeration loop consists of a gas turbine driving a direct connected centrifugal nitrogen cycle compressor. The main cycle compressor receives the nitrogen refrigerant gas at approximately 5.5 atm. and compresses it to 27 atm. After appropriate aftercooling, this gas is further compressed by driven expander loading compressors to an ultimate pressure of approximately 45 atm. Thus, the refrigerant gas, compressed and aftercooled, is delivered to the cold box at this pressure where it begins the exchange process in plate fin exchangers. The pressures have been selected so that commercially available plate fin exchangers can be utilized throughout. After an initial exchange against returning refrigerant gas, the refrigerant stream is expanded through a nitrogen expander and then reheated, for curve bending purposes, before final expansion for counter flow through the exchangers. Part of this low temperature nitrogen stream is liquefied for expansion through a Joule-Tompson valve and delivered to a holdup exchanger from which it may be either withdrawn or used to recondense boil-off from the main storage tank. The balance of the nitrogen refrigerant stream is exchanged against incoming process gas and ultimately returned to the main nitrogen cycle compressor inlet. Makeup nitrogen gas from the pure methane column is returned to this process stream on its route to the cycle compressor.

Returning now to the main process stream, it is initially cooled against returning nitrogen to a temperature which will knock out C_5 plus components which are delivered to the gas turbine fuel supply stream. By eliminating these components at this level, we have determined that the remaining constituents will not solidify at temperatures well below temperatures encountered in our process.

After further temperature reduction, the main stream passes through a nitrogen stripping column where all of the nitrogen and a portion of the methane is separated and delivered to a nitrogen-methane separation column. In this nitrogen-methane column, 99 percent plus pure methane is removed as a liquid and delivered to pure methane storage. The top of this column delivers pure nitrogen gas which is returned to the refrigerator as make up. Returning our attention to the bottom of the nitrogen stripping column, this gas is returned to the exchangers for further temperature reduction and liquefaction, and for final delivery to the LNG storage tank.

You will recall that one important feature of the cycle is elimination of cold blowers. This is done very simply by recondensing the main storage tank boil off vapors against liquid nitrogen and by collecting this condensed liquid through a simple liquid eductor and delivering it to the LNG storage tank. Motive power for the eductor is supplied by the main LNG liquid stream which handles the entire boil off gas stream without the introduction of a dynamic piece of equipment. With this system there is no "weathering" of the stored liquid product because there is no net removal of gas from the main LNG storage tank. As you know, the boil off from a LNG storage tank tends to be high in nitrogen and in pure methane, tending to accumulate heavy hydrocarbon components in the storage tank.

Another very important feature of the system utilizing the nitrogen cycle is the fact that the final main stream of LNG product to be delivered to the LNG storage tank is subcooled prior to its pressure letdown. The importance of this is the fact that upon pressure letdown for introduction into the storage tank, there is no gaseous flash from the main product stream. This again is a very important feature in maintaining product composition because with other processes this flash again is particularly rich in nitrogen and methane and tends to increase the concentration of heavies retained in the storage tank.

For those periods of time when the storage tank is full and the main refrigeration cycle is not operative, a boil off recondensation system can be used although this is not shown on the simplified flow diagram. This system simply utilizes some of the main components and continually recondenses main tank boil off, maintaining constant composition in the storage tank. During periods of operation, excess gaseous nitrogen from the cycle is stored in a high pressure bottle system at approximately 45 atm. for use as a future purge medium and as a refrigerant. Recognizing that a LNG peak shaving plant will be inoperative during approximately 100 days of the year and that the refrigeration cycle may be called upon to start and stop frequently during this period of time for tank topping off purposes, the use of a refrigerant which can be stored at ambient conditions had many advantages. With this reserve of gaseous dry nitrogen, not only is the refrigerant gas available in a convenient form, but nitrogen gas is available for derime and purging purposes. This nitrogen is also available for fire fighting, storage tank insulation space, and cold box purge purposes as well as pneumatic instrument services.

In terms of operating requirements and utilities, the plant is quite undemanding. Natural gas is the major need. For a gross liquefaction rate of 200,000 standard gaseous cubic meters per day, 230,000 standard gaseous cubic meters of feed gas must be supplied. Bear in mind that this excess provides for the multiple services of cleanup system regeneration, main cycle compressor power supply and through nitrogen separation, refrigerant supply and purge gas supply, as well as liquid nitrogen withdrawal. Electrical power is also required for various auxiliary pumps, compressor, control and heating services; and the total connected load, excluding revaporization services, is 800 kw. For purposes of simplicity, as requested by the program committee, the flow diagram does not include the various auxiliaries needed for a complete plant. Not shown are a feed gas pressure booster compressor, whose load will vary depending upon gas supply pressure, a main storage tank boil off compressor, to be used to compress normal tank boil off to distribution main pressure during periods of time when the storage tank is full and the main refrigeration cycle is inoperative, and the revaporization and sendout system.

In summary then, this plant now under construction will have no effluent or odorant disposal problems since these items are consumed by the prime mover, will have no calorific adjustment problems since the storage tank composition remains constant, will have no refrigerant supply problems or purge gas supply problems since provision is made for these services as a part of the overall cycle consideration. With the use of centrifugal machinery and its consequent moderate pressures and the use of commercially available exchangers throughout, we believe that operational problems will be minimal, resulting in high plant availability, high on stream factors and hence, extremely economical operation.

FIG. 1 TEMPERATURE – ENTHALPY PLOT
FOR COLD EXCHANGER
180 ATM NITROGEN

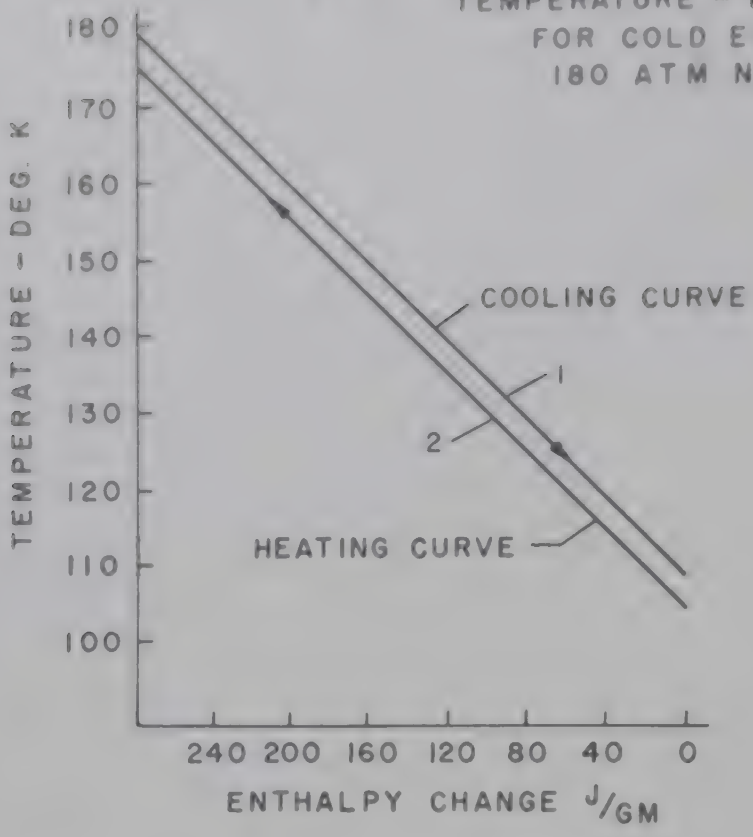


FIG. 2 TEMPERATURE – ENTHALPY PLOT
FOR COLD EXCHANGER
45 ATM NITROGEN

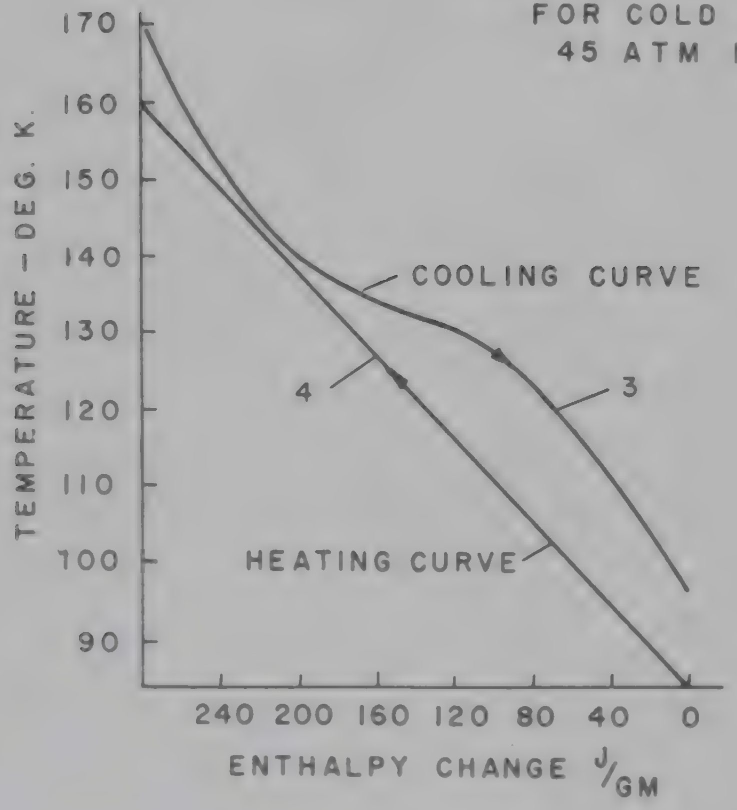
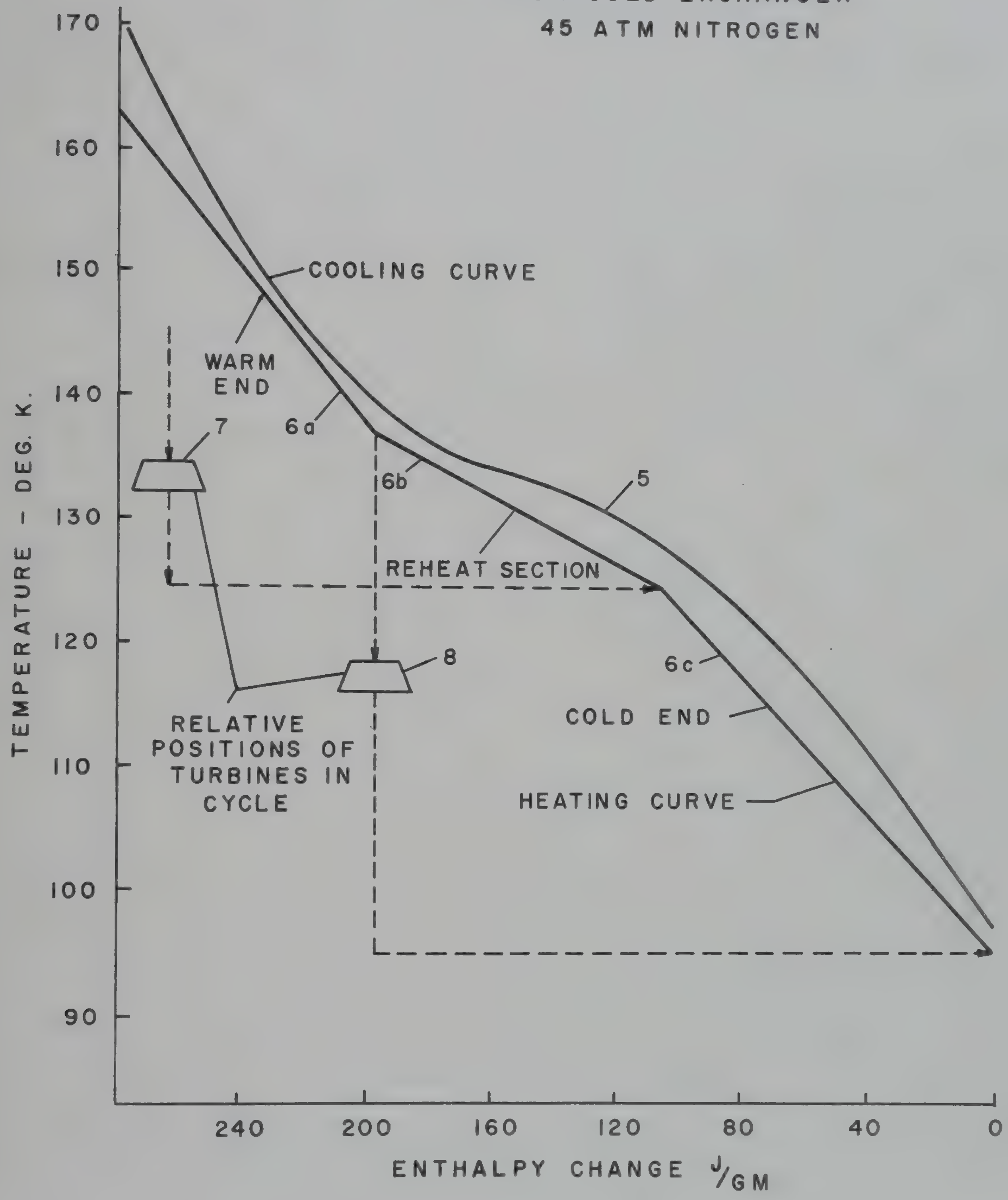


FIG. 3

TEMPERATURE — ENTHALPY PLOT
FOR COLD EXCHANGER
45 ATM NITROGEN



AIRCO/BOC CPC PATENT APPLIED FOR

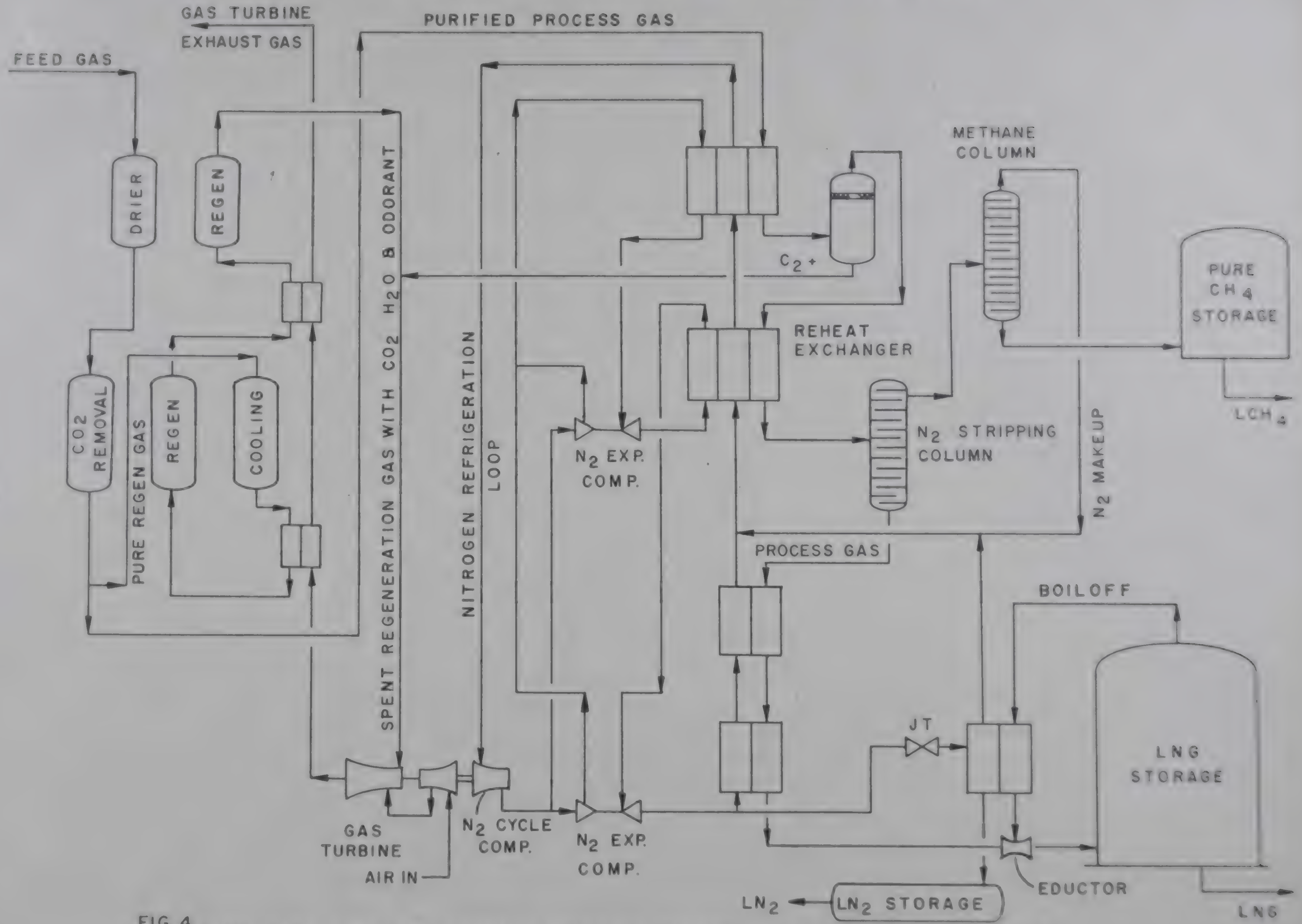

GAS TURBINE
EXHAUST GAS
FEED GAS
PURIFIED PROCESS GAS
DRIER
REGEN
CO2 REMOVAL
PURE REGEN GAS
REGEN
COOLING
SPENT REGENERATION GAS WITH CO2 H2O & ODORANT
NITROGEN REFRIGERATION LOOP
N2 EXP. COMP.
C2 +
REHEAT EXCHANGER
METHANE COLUMN
PURE CH4 STORAGE
LCH4
N2 STRIPPING COLUMN
N2 MAKEUP
PROCESS GAS
BOILOFF
JT
LNG STORAGE
LNG
GAS TURBINE
AIR IN
N2 CYCLE COMP.
N2 EXP. COMP.
EDUCTOR
LN2
LN2 STORAGE

FIG 4
SIMPLIFIED NITROGEN REHEAT EXPANSION NATURAL GAS LIQUEFACTION CYCLE

Some Considerations on Peakshaving with Dutch Natural Gas

Dipl.-Phys.W.Förg

By 1971 Southern Germany will be connected by pipeline to the gasfields of the Dutch region Groningen. For several South German gas suppliers the question of peakloading with this natural gas presents problems.

Taking, for example, the Technische Werke of Stuttgart, several problems of peakshaving plants will be discussed. Fig.1 shows the block diagram of the proposed plant, whose most important part is a LNG storage tank with a capacity of 35 000 m^3, which corresponds to a gas volume of 2o ooo ooo Nm^3. The LNG will be stored at a temperature of about 110°K. The storage tank is designed for 4oo mm water gauge pressure above atmospheric and a boil-off-rate of o.o5 % of the tank capacity per day.

The following alternatives are considered:

1. A double-walled metal tank, which is surrounded by a concrete shell with earth-fill.
2. An earth-fill supported concrete tank with a closed, inner metal tank.

The insulation space should, in both cases, on the grounds of safety, be purged with nitrogen.

For the reevaporation three units, each having a capacity of 25 ooo Nm^3/h, are planned. These units evaporate the LNG, compressed by submerged pumps to about 21 atmospheres and warm it up to ambient temperature.

These evaporators consist of special tube bundles, which are immersed in a warm water bath, which is directly heated by means of an immersed flame.

The plant is fed with Dutch gas with the following approximate composition:

	mole %
He	0.04
CO_2	0.80
N_2	14.30
O_2	0.01
CH_4	81.59
C_2H_6	2.60
C_3H_8	0.40
$i-C_4H_{10}$	0.08
$n-C_4H_{10}$	0.08
$i-C_5H_{12}$	0.02
$n-C_5H_{12}$	0.02
C_{6+}	0.06
	100.00

Temperature		30°C
Pressure		21 ata
C.V.	about	8400 kcal/Nm^3
rel.Density		0.64
Dewpoint		0°C at 5 ata

Before the gas liquefaction CO_2 and H_2O must be removed, as they would lead to blockages in the liquefaction plant. The CO_2-removal is achieved by means of a MEA-wash and the drying by means of two interchangeable molecular-sieve adsorbers.

During the liquefaction the heavy hydrocarbons are separated off within the precooling system of the plant, as these components (e.g.benzene) can also lead to blockages.

In the low temperature section of the plant it is advisable to remove nitrogen, as this would lower the boiling point of the LNG from about 110°K to about 91°K. The boil-off from the LNG storage tank, which results from insulation heat leakage, would then consist of about 88 mole % N_2 and about 12 mole % CH_4. Thus,without continual reliquefaction,the composition of the LNG would vary as a strong function of the storage time.

For the N_2-removal two alternatives were considered.

1. 3833.8 Nm^3/h of LNG with a N_2-content of max. o.5 mole % are added to the storage tank. At the same time, 2o83.4 Nm^3/h of gas with a C.V., which is at most 5 % below the feed C.V. are added to a large, existing gasholder. 5oo Nm^3/h of N_2 are separated off.

2. 3833.3 Nm^3/h of LNG with a N_2-content of max. o.5 mole % are added to the storage tank. Only the boil-off from the storage tank is allowed to be added to the existing gasholder. 5oo Nm^3/h N_2 are separated off.

Fig. 2 shows the process to separate N_2 by alternative 1. In this process the NG is cooled down, after separating off the heavy hydrocarbons, so that the N_2-content is about 5 mole % in the liquid. The remaining gas is separated in Separator 1. It is then used to heat the Lower Section 2 of the high-pressure part of a double column. The bottom product from Column 3 also contains about 5 mole % of N_2. It is mixed with the already-subcooled liquid from the aforementioned Separator 1 and further subcooled. The N_2-rich gas from Separator 1, after heating Reboiler 2 is cycled back to the precooling section of the plant. It is then cooled down in Heat Exchanger 6 by separation products, which need to be warmed up. Then it is expanded in Pressure Column 3. The Condenser 4 of Pressure Column 3 - as is already known from air-separation plants - serves as an evaporator for the Upper Column 5, which operates under low pressure. The separated N_2, which comes from Condenser 4 is expanded and cools down the LNG from Evaporator 4 of the Upper Column 5 in Heat Exchanger 7. Then the N_2 is further warmed up in the liquefaction unit.

At the top of Upper Column 5, the subcooled liquids, which contain about 5 mole % of N_2 are expanded. This N_2 is stripped out. The gas removed from the top of the column contains about 18 mole % of N_2. By mixing with the heavy hydrocarbons in the precooling section the C.V. of this gas is adjusted to that of the feedgas. Then it is added to the existing Gasholder 9. The N_2-free LNG leaves from the bottom of the upper column is further cooled in Subcooler 7 and is added to the LNG Storage Tank 8.

By this method the final cooling is achieved by expansion of LNG. This expansion is not achieved by means of a simple valve arrangement, but by coupling with the double column system.

Because of this, the pressure-energy of a part of the feedgas can be utilized by using Gasometer 9 and simultaneously separating N_2 from the liquefied product.

Fig.3 shows a process, by which no gas from the liquefaction plant may be added to the gasholder. In this case also, a double column is used, which works in another manner. The feedgas heats Evaporator 1 of the Pressure Column 2, is cooled in the precooling Section and in Heat Exchanger 5 and then expanded in Pressure Column 2. In the Condenser 3 of the pressure column the reflux liquid for Upper Column 4 is produced. This liquid is subcooled in Heat Exchangers 6 and 7 and expanded in the upper column. Condenser 3 has an inertgas-remover, by which a He build-up may be avoided.

As opposed to the process shown in Fig.2 the pure N_2 here is produced in the upper column. It leaves the top of this column and is warmed up in Heat Exchangers 7, 6, 5 and finally in the precooling section. The bottom product of the pressure column is subcooled in the precooling section and Heat Exchangers 5 and 6 and is then expanded in the upper column. There it is separated into pure N_2 and N_2-free CH_4. The liquid CH_4 is taken from Evaporator 3 of the upper column, is cooled down in Heat Exchanger 6 and is fed into the LNG Storage Tank 8. The final cooling in this process is produced by a liquid mixture of N_2 and CH_4. This mixture is taken from the pressure column, subcooled in Heat Exchanger 6, expanded and evaporated in Heat Exchangers 6 and 5, before being warmed up in the precooling section and recompressed in the feedgas.

The energy consumption of the process shown in Fig.3 is higher than that of the process shown in Fig.2 because in Fig.2 no compression work for the recycle to the feedgas is required.

Fig.4 shows the two-phase-equilibrium of the system N_2-CH_4. This diagram shows at a glance, which temperatures can be reached at given compositions and pressures. The necessary conditions for the condenser of the pressure column (evaporator of the upper column) can be easily calculated.

Certainly one can use other methods of N_2-removal with double columns, single columns or simple separators. For N_2-rich natural gases, as is the case with the Dutch gas, it is certainly economic to use the somewhat more complicated double-column system.

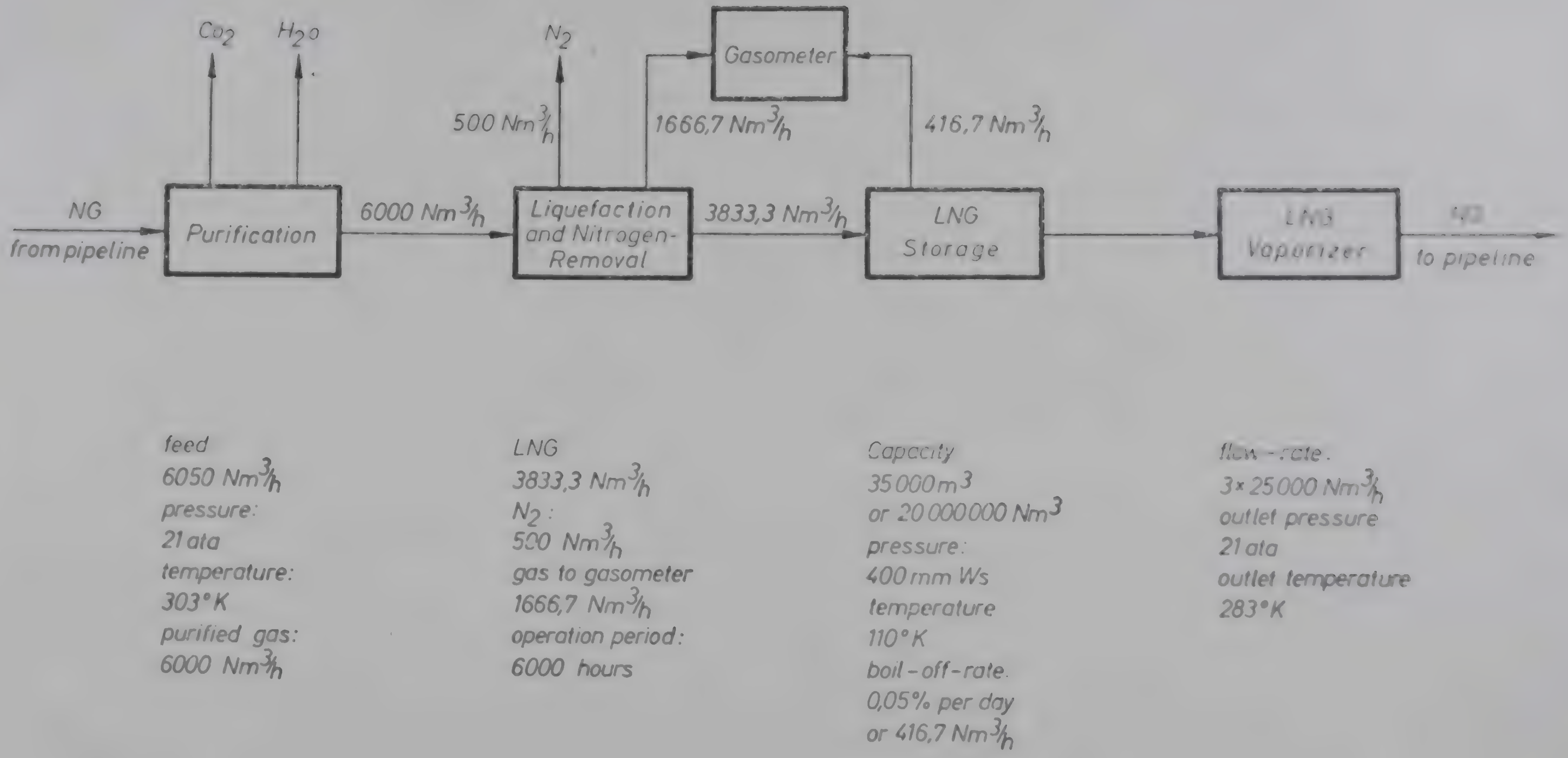

CO_2
H_2O
NG
from pipeline
Purification
6000 Nm^3/h
N_2
500 Nm^3/h
Liquefaction and Nitrogen-Removal
1666,7 Nm^3/h
Gasometer
3833,3 Nm^3/h
416,7 Nm^3/h
LNG Storage
LNG Vaporizer
NG
to pipeline
feed
6050 Nm^3/h
pressure:
21 ata
temperature:
303°K
purified gas:
6000 Nm^3/h
LNG
3833,3 Nm^3/h
N_2:
500 Nm^3/h
gas to gasometer
1666,7 Nm^3/h
operation period:
6000 hours
Capacity
35000 m^3
or 20000000 Nm^3
pressure:
400 mm Ws
temperature
110°K
boil-off-rate.
0,05% per day
or 416,7 Nm^3/h
flow-rate.
3×25000 Nm^3/h
outlet pressure
21 ata
outlet temperature
283°K

Fig.1. Peak shaving plant og TW Stuttgart, Germany

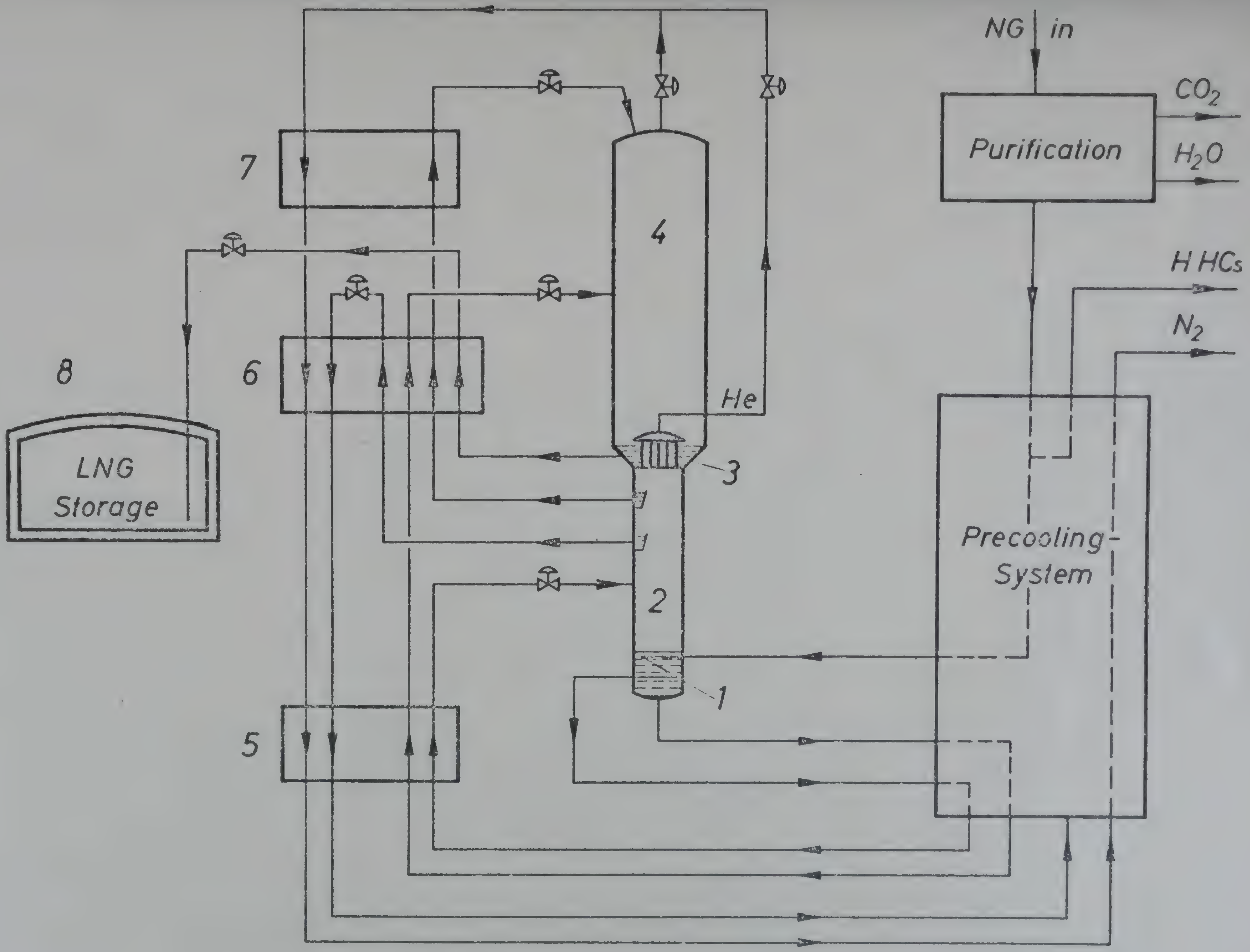

NG in
Purification
CO₂
H₂O
H HCs
N₂
Precooling-System
He
1
2
3
4
5
6
7
8
LNG Storage

Fig.2. Nitrogen removal in a LNG plant

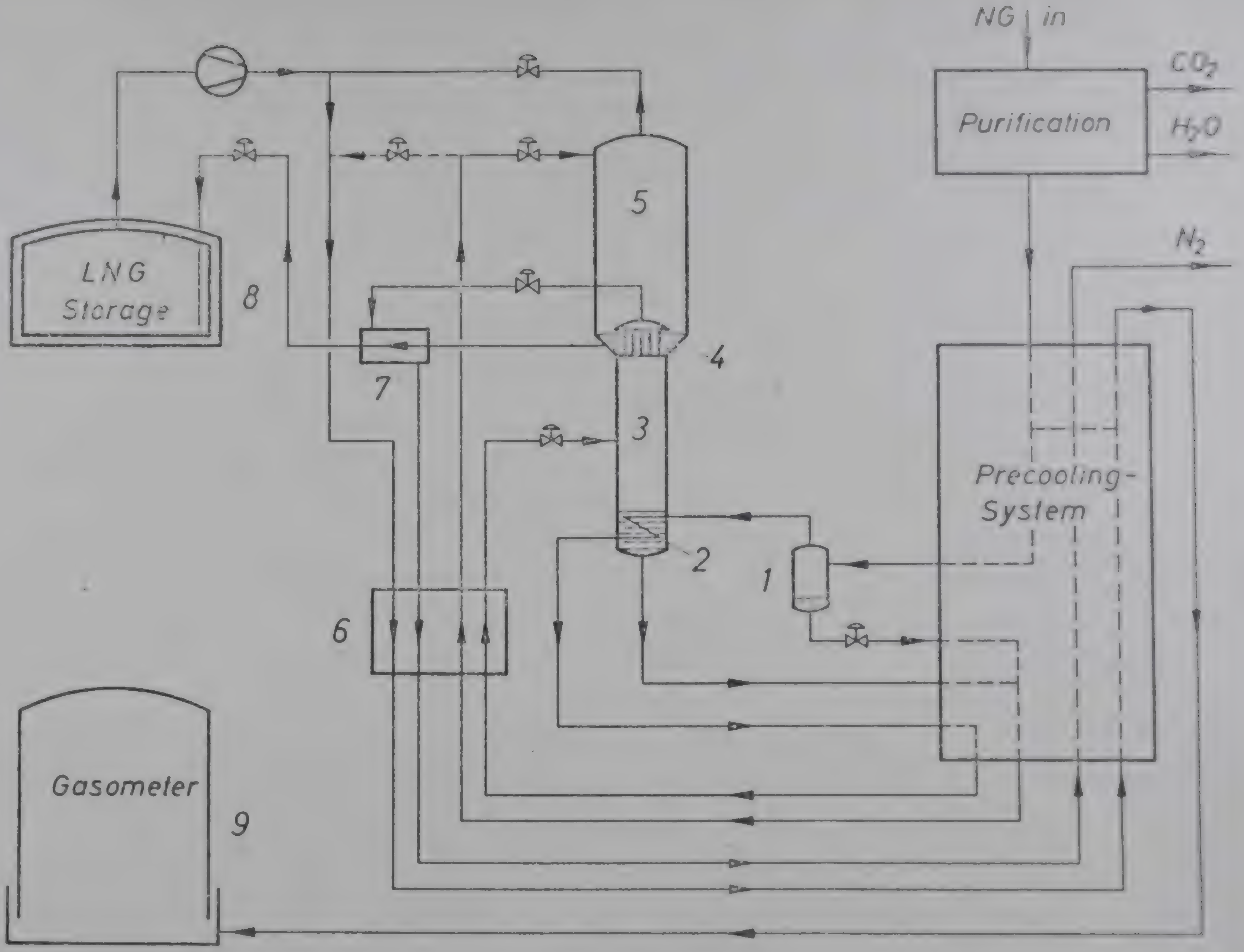

NG in
Purification
CO_2
H_2O
N_2
Precooling-System
LNG Storage
Gasometer
1
2
3
4
5
6
7
8
9

Fig.3. Nitrogen removal in a LNG plant

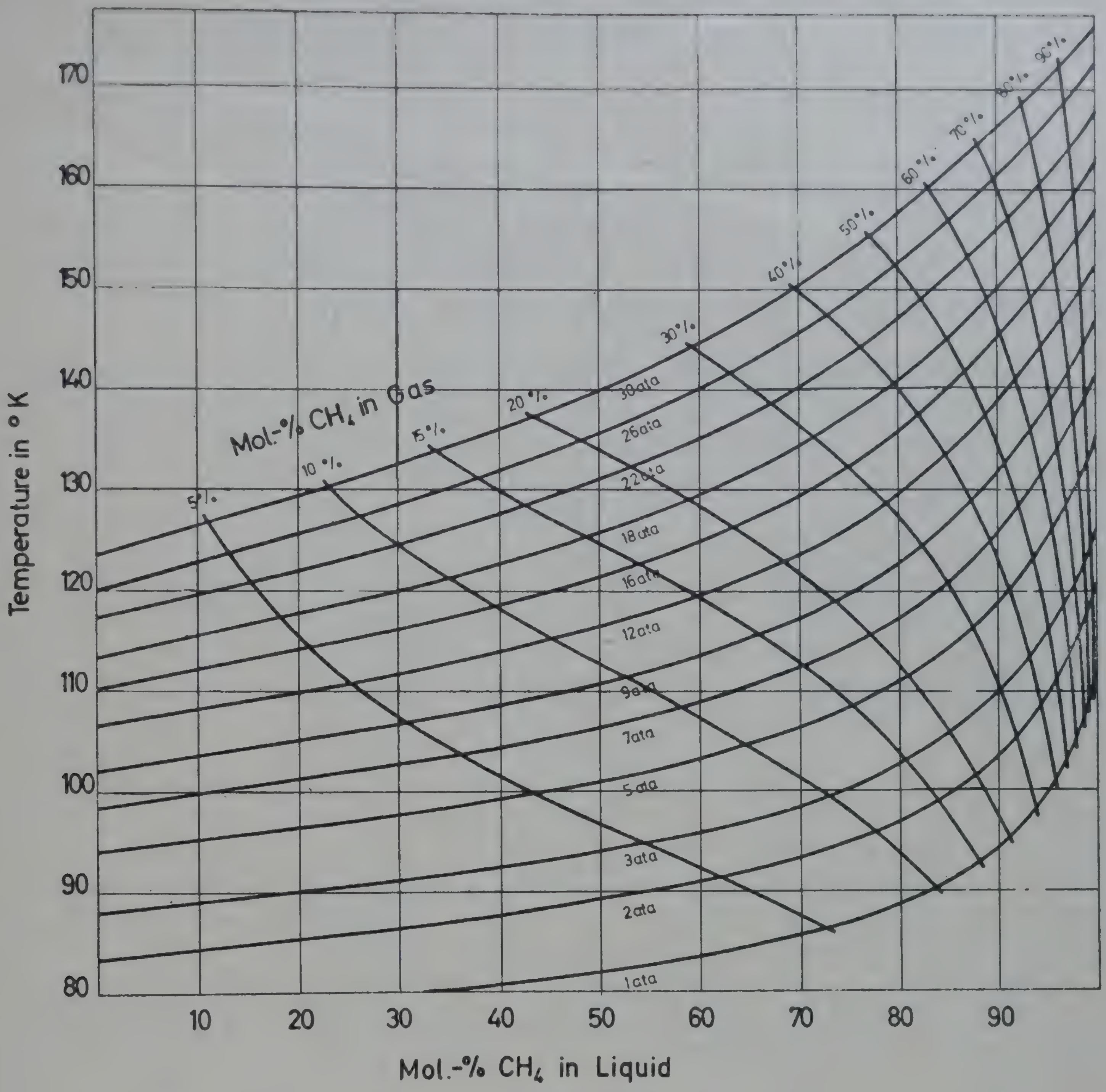


Fig.4. Two phase equilibrium N_2 CH_4

DEVELOPPEMENT DU CYCLE A CASCADE INCORPOREE POUR LA LIQUEFACTION DU GAS NATURAL

By B. Darredeau

I. INTRODUCTION

De nombreuses communications ont déjà souligné l'évolution récente et les perspectives d'avenir considérables de l'industrie du G. N. L. dont les deux principales applications jusqu'ici sont le transport intercontinental en grandes quantités et l'écrétage des pointes de consommation dans les usines de "peak-shaving".

L'extension du champ des applications du G. N. L. est subordonnée à l'évolution des techniques de liquéfaction, de stockage et de transport. Précisément, dans ce domaine, un effort considérable d'étude et de recherche a été entrepris, effort justifié par les perspectives prometteuses de cette jeune industrie.

En particulier, les techniques de liquéfaction ont progressé sous l'impulsion de Sociétés ayant l'expérience des basses températures. C'est ainsi que la Société TEAL, créée pour exploiter les brevets d'Air Liquide et de Technip, a mis au point le cycle à cascade incorporée dont l'intérêt n'est plus à démontrer et qui a déjà fait l'objet d'un certain nombre de communications par des spécialistes.

Il n'est cependant pas superflu de rappeler succinctement, pour la clarté de l'exposé, le principe de ce procédé.

II. DESCRIPTION DU CYCLE A CASCADE INCORPOREE (fig. 1)

Le cycle à cascade incorporée comporte un fluide réfrigérant unique composé d'un mélange d'hydrocarbures et d'azote.

Ce fluide réfrigérant, comprimé à 30 ata, subit une succession de condensations partielles, le liquide formé étant extrait à chaque étage de condensation et revaporisé en basse pression dans l'étage suivant pour y fournir les frigories nécessaires.

A chaque étage, l'extraction du liquide formé appauvrit le réfrigérant en hydrocarbures lourds si bien que la vapeur restante, de plus en plus légère, se condense et se vaporise à des niveaux de température de plus en plus bas, permettant ainsi d'atteindre de proche en proche la température de condensation du gaz naturel puis sa température de sous-refroidissement final aux environs de -162°C. En effet, dans ce procédé, le gaz naturel liquéfié est également sous-refroidi sous pression jusqu'à sa température d'ébullition à la pression atmosphérique. Il peut alors être envoyé directement au stockage sans pompe de transfert et sans qu'il ne se produise aucun flash lors de la détente à la pression atmosphérique. Ce sous-refroidissement complet est possible grâce à la présence dans le réfrigérant d'une certaine quantité d'azote dont la température de vaporisation en basse pression est inférieure à la température d'ébullition du G.N.L.

Nous n'insisterons pas longuement sur les multiples avantages que peut présenter ce procédé par rapport aux autres cycles de liquéfaction :

- Le rendement thermodynamique est bon puisqu'il est comparable à celui de la cascade classique

- Un seul compresseur de cycle est nécessaire. C'est d'ailleurs la seule machine tournante puisqu'aucune pompe de transfert de G.N.L. n'est prévue. La concentration de la puissance sur un seul compresseur permet d'utiliser une machine plus importante dont le rendement est meilleur

- Le nombre des échangeurs de cycle est réduit (4 ou 5 en général)

- Les tuyauteries de liaison sont peu nombreuses. Moyennant une disposition particulière des échangeurs, il est possible de réaliser un ensemble prémonté en usine, ce qui réduit particulièrement le coût du montage sur le site.

- Le réfrigérant est constitué à partir de fractions lourdes extraites du gaz naturel, moyennant des petites colonnes de distillation. Il n'est pas nécessaire d'approvisionner et de stocker des produits purs coûteux.

Pour l'expérimentation et la mise au point de ce procédé, l'Air Liquide a construit avec Gaz De France une unité pilote dont on trouvera le schéma en annexe. (fig. 2).

La principale particularité de ce schéma est le fait que les circuits de gaz naturel et de gaz de cycle sont mélangés à partir du 3ème échangeur. Il faut donc extraire à l'extrémité froide de l'appareil un liquide ayant la même composition que le gaz naturel traité afin de conserver une quantité et une composition constantes de réfrigérant dans l'appareil. Le rôle des colonnes de distillation est de permettre l'ajustement de cette composition.

Les caractéristiques du pilote sont les suivantes :

Capacité	1 000 Nm3/h
Puissance	600 cv (chiffre élevé dû essentiellement au faible rendement du compresseur de cycle)
Cycle	5 500 Nm3/h entre 6 bars et 30 bars

III. CALCUL DU PROCEDE

A. CALCUL THERMODYNAMIQUE

1°) Méthode de calcul

Pour la mise en œuvre de ce procédé, la première difficulté à laquelle on se heurte est le calcul thermodynamique, c'est-à-dire la détermination de toutes les grandeurs caractéristiques concernant le réfrigérant :

- composition
- pressions
- températures
- débits

Les données dont on dispose sont généralement les suivantes :

- composition, pression, température et débit du gaz naturel à liquéfier et, éventuellement, composition imposée pour le liquide produit

- température de l'eau de réfrigération ou, éventuellement, de l'air ambiant.

A partir de ces données, on effectue un calcul de condensation sur la machine électronique pour déterminer, en fonction de la température, les équilibres liquide-vapeur. Grâce à ces résultats, on peut connaître la courbe d'évolution de l'enthalpie du gaz naturel en fonction de la température.

Le nombre élevé des paramètres à déterminer rend particulièrement délicat le calcul. En pratique, l'expérience permet de fixer à priori un certain nombre de ces paramètres à des valeurs que l'on sait raisonnables sinon optimales. (fig. 3).

Dans chaque échangeur, les besoins frigorifiques en fonction de la température résultent de la somme des frigories à fournir sur les trois fluides suivants :

- gaz naturel
- gaz de cycle en condensation
- liquide de cycle à sous-refroidir

En faisant la somme des enthalpies de ces trois fluides à chaque température, on obtient la courbe de refroidissement dans l'échangeur considéré.

Il reste alors à calculer la courbe de vaporisation du fluide de cycle correspondant, dans cet échangeur.

L'ensemble des deux courbes ainsi obtenues :

- courbe de refroidissement
- courbe de vaporisation

constitue le diagramme d'échange à partir duquel on peut déterminer l'écart de température le long de l'échangeur.

Il se peut que l'écart de température ainsi obtenu soit négatif en certains points. Cela signifie que le bilan n'est pas correct. On doit alors modifier un ou plusieurs paramètres et recommencer le calcul.

En réalité, tout ce calcul est exécuté assez rapidement parce qu'il est possible d'introduire toutes les données dans la machine et d'obtenir les calculs d'équilibres sur tous les fluides en une seule fois grâce à un programme spécialement mis au point à cet effet. D'autre part, l'approche est relativement rapide, moyennant une bonne connaissance de l'influence de chacun des paramètres.

Dès que les bilans en tous points sont corrects, on peut réaliser l'optimisation thermodynamique qui consiste, en particulier, à obtenir une répartition régulière des écarts de température sur les échangeurs, sachant que des écarts trop faibles entraînent un gaspillage de surface et, inversement, que des écarts trop grands représentent une dégradation d'énergie.

2°) Données de base

Bien entendu, la réalisation de ces calculs suppose la connaissance d'un certain nombre de données physiques. Les plus importantes sont les cœfficients d'équilibre qu'il est nécessaire de connaître avec une grande précision. En effet, la plus grande partie de la chaleur échangée, dans un tel procédé, est transférée sous forme de chaleur latente de vaporisation. Or, la quantité de frigories disponible à chaque température dépend de la quantité de liquide qui est vaporisé à cette température, c'est-à-dire de l'équilibre liquide-vapeur du fluide considéré.

Si une erreur sur les cœfficients d'équilibre entraîne par exemple une erreur de 1°C tout au long de la courbe de vaporisation d'un fluide, et par conséquent sur l'écart de température moyen de l'échangeur considéré, il en résulte une différence de surface d'échange qui, compte tenu des petits écarts de températures adoptés sur les échangeurs, est très importante.

On conçoit donc que la précision requise pour le calcul des équilibres est très grande.

C'est pourquoi un effort particulier a été consenti dans la détermination de ces données d'équilibre.

Dans une première étape, un travail théorique a été entrepris pour leur détermination par le calcul, en fonction de la température, de la pression et de la composition du mélange considéré. Les valeurs théoriques ont été comparées à des valeurs expérimentales connues dans certains cas simples, par exemple pour des binaires, ou à des valeurs issues de la littérature. Certains ajustements ont pu ainsi être faits. On a alors pu effectuer des calculs de cycle qui ont permis en particulier de dimensionner le pilote de Nantes. Deux sortes de vérifications expérimentales ont, par la suite, été entreprises :

- d'une part, une vérification industrielle : on s'est assuré, a posteriori, que les calculs ayant servi pour le dimensionnement du pilote étaient confirmés globalement par la production de celui-ci qui s'avérait conforme à la valeur escomptée

- d'autre part, en laboratoire, on s'est livré à un certain nombre de mesures de cœfficients d'équilibres sur des mélanges dont la composition correspondrait à celle des fluides de cycle calculés.

Il est ainsi apparu que, si les calculs étaient confirmés par le résultat global obtenu, il y avait cependant quelques légères erreurs qui, dans l'ensemble, se compensaient mais qui amenaient à une répartition différente des surfaces d'échange.
En fait, on démontre par le calcul qu'il est relativement facile de corriger ces erreurs par une modification légère de la composition du gaz de cycle. C'est ainsi que, l'appareil étant construit, on pourra fonctionner dans des conditions différentes du design et la composition optimum du gaz de cycle sera alors ajustée expérimentalement en partant de la composition de calcul.

B. DIMENSIONNEMENT DES ECHANGEURS

1°) Description des échangeurs utilisés (fig. 4)

Les échangeurs que l'on utilise le plus souvent dans les appareils à cascade incorporée sont des échangeurs bobinés.

Autour d'un noyau central qui constitue l'axe vertical de l'échangeur, des tubes sont enroulés par couches successives. L'angle des tubes, par rapport à l'horizontale, est très faible (de l'ordre de 5 à 10°C). Les couches des tubes sont maintenues écartées l'une de l'autre par un certain nombre de cales verticales placées sur toute la périphérie de chaque couche.

A chaque extrémité de l'échangeur, les tubes aboutissent dans une boîte à laquelle ils sont reliés de façon étanche. Il peut y avoir plusieurs boîtes, chacune d'elles correspondant à un faisceau de tubes différent et, éventuellement, à un fluide différent. Dans les tubes, circulent les fluides en haute pression. A l'extérieur des tubes, le fluide en basse pression se réchauffe. Ce type d'échangeur, à courants antiparallèles, permet d'obtenir des écarts de températures très faibles entre les fluides échangeant de la chaleur. D'autre part, il est possible de réaliser des échangeurs de très grosse taille dont l'intérêt est bien évident pour les usines de liquéfaction de grande capacité.

2°) Précautions technologiques

Après avoir effectué les calculs thermodynamiques permettant de déterminer à chaque température l'équilibre liquide-vapeur d'un fluide, il reste à réaliser un équipement tel que cet équilibre soit effectivement obtenu en tout point. En effet, si il se produit une séparation du liquide et de la vapeur au cours de la vaporisation le liquide étant plus lourd se vaporise à un niveau de température plus élevé. C'est exactement l'inverse dans le cas de la condensation, si bien que, dans les deux hypothèses, l'écart moyen obtenu est plus faible que celui calculé et les performances sont inférieures pour une surface d'échange donnée. C'est pourquoi il est nécessaire de réaliser un mélange intime du liquide et de la vapeur en tout point.

Dans ce but, on maintiendra dans l'échangeur une vitesse d'écoulement suffisamment élevée qui, par ailleurs, doit favoriser l'échange de chaleur. Cependant, on est limité dans ce sens par la valeur de perte de charge admissible dans l'échangeur, spécialement en basse pression. En effet, un accroissement de la perte de charge de 100 g/cm2 sur la ligne d'échange en basse pression correspond à une dépense d'énergie supplémentaire de 1,5% environ.

D'autre part, il est indispensable de répartir correctement sur toute la section de l'échangeur, le liquide à vaporiser. C'est pourquoi un dispositif spécial de distribution du liquide a été mis au point.

La détermination de la vitesse minimum au-dessous de laquelle le liquide n'est pas correctement entraîné et peut se séparer du gaz a été faite pour l'air et l'eau grâce au dispositif suivant :

Entre deux plaques de plexiglass parallèles, des tubes droits horizontaux disposés en rangées verticales reproduisent une partie de l'obstacle présenté par le bobinage à l'écoulement du gaz. L'air injecté au bas du dispositif le traverse de bas en haut avec une vitesse variable. L'eau injectée également en proportions variables est entraînée en totalité quand la vitesse de l'air atteint 5 m/sec.

En tenant compte des densités du liquide et du gaz par rapport à l'eau et à l'air et, plus précisément du facteur :

$$\sqrt{\frac{\rho_L - \rho_v}{\rho_v}}$$

il a été possible de déterminer la vitesse correspondante pour les hydrocarbures. Afin d'obtenir une bonne souplesse tout en tenant compte des impératifs concernant les pertes de charge, la vitesse d'écoulement adoptée au point nominal dans les échangeurs est plus élevée que celle qui est déterminée par cette méthode. D'autre part, sur le même dispositif, on a mesuré les pertes de charge obtenues pour différentes proportions de liquide entraîné et on les a comparées à celles que l'on obtenait pour le gaz seul.

C'est à partir de ces différentes mesures que les échangeurs du pilote ont été définis.. Des essais de fonctionnement à débit réduit ont montré que la souplesse obtenue permettait une bonne opération à 50%. D'autre part, les mesures de pertes de charge ont confirmé les conclusions de ces essais.

Ce problème de la distribution du liquide à l'entrée de chaque échangeur et de son mélange intime avec le gaz au cours de la vaporisation a fait l'objet d'essais plus récents afin de contrôler les hypothèses qui ont été faites pour l'extrapolation nécessaire lors du passage de la taille du pilote à celle d'une usine de liquéfaction de grande capacité. Ces essais ont montré que le même dispositif restait parfaitement valable, moyennant certaines précautions en ce qui concerne la disposition des injections.

3°) Méthodes de dimensionnement

Le dimensionnement des échangeurs cryogéniques utilisés dans le procédé "à cascade incorporée" résulte d'une optimisation qui tient compte :

- de l'échange thermique imposé à l'élément
- des régimes d'écoulement des fluides
- de considérations et d'impératifs technologiques et économiques

a) Echange thermique imposé à l'élément

Celui-ci est fixé au thermicien par l'ingénieur de procédé qui fournit les données suivantes :

- quantités de chaleur échangée sur chaque fluide
- écarts de température entre fluides
- débit, état, caractéristiques physiques de chaque fluide

b) Régimes d'écoulement

Un dimensionnement approximatif est effectué en prenant par exemple des moyennes sur les écarts de températures, des vitesses de circulations de fluide et des cœfficients d'échange dont l'ordre de grandeur est connu. On calcule les cœfficients d'échange et pertes de charge sur chaque fluide en plusieurs points de l'échangeur et on rectifie alors le dimensionnement. On opère par itération jusqu'à obtenir la convergence.

Il y a lieu de tenir compte de certains impératifs de vitesse dans ce calcul.

c) Impératifs technologiques

- les échangeurs bobinés les plus gros ne peuvent dépasser un certain diamètre
- une bonne proportion diamètre-hauteur doit être respectée
- il y a lieu de tenir compte du rayon minimum de cintrage des tubes
- l'échangeur optimum sera celui qui, pour des performances données, aura la surface d'échange minimum et le nombre de tubes minimum, moyennant des pertes de charge acceptables.

Nous avons donc été amenés à étudier de très près le problème des cœfficients d'échange et pertes de charge en régime diphasique liquide-vapeur, ce qui est le cas général pour les échangeurs cryogéniques du procédé "à cascade incorporée".

La première partie de ce problème consiste à déterminer, pour un dimensionnement approximatif, les vitesses d'écoulement du liquide et de la vapeur, puis à calculer les pertes de charge.

Pour ce faire, on passe en général par l'intermédiaire d'équations de pertes de charge du genre de l'équation de Fanning

$$\frac{\Delta P}{L} = k \rho V^2$$

k = facteur de friction dépendant des parois et du nombre de Reynolds

ρ = masse volumique

v = vitesse

On peut écrire, en considérant un conduit, que pour une faible longueur de ce conduit, les pertes de charge de chaque phase sont égales, ce qui donne les sections de circulation de chaque phase (Méthode de Gloyer)

$$S_L = SF\ell = S \left[\frac{1}{\frac{W_v}{W_L} \times \left(\frac{\rho_L}{\rho_v}\right)^{0,5} \times \left(\frac{\mu_v}{\mu_L}\right)^{0,1} + 1} \right]$$

et $S_v = S - S_L = S\ (1 - F\ell)$

On peut encore utiliser des méthodes du genre de celles de Martinelli Nelson, Martinelli Lockhart, Chenoweth Martin, etc... qui relient les pertes de charge en diphasique, aux pertes de charge en monophasique. (Cette perte de charge en monophasique est calculée sur les caractéristiques de la phase considérée dans toute la section, avec son débit massique propre, ou le débit total).

Martinelli Nelson

$$\frac{\Delta P_{TP}}{\Delta P_\ell} = \emptyset^2_{\ell tt}$$

où $\emptyset^2_{\ell tt}$ est donné par un abaque en fonction de X_{tt} (paramètre de Martinelli Nelson)

$$X_{TT} = \left(\frac{W_L}{W_v}\right)^{0,9} \times \left(\frac{\rho_v}{\rho_L}\right)^{0,5} \times \left(\frac{\mu_L}{\mu_v}\right)^{0,1}$$

Le même genre de relation existe entre ΔP_{TP} et ΔP_v.

Chenoweth Martin

Un abaque, fonction du taux de vide :

$$LVF = \frac{1}{1 + \frac{\text{taux poids vapeur}}{\text{taux poids liquide}} \times \frac{\rho_L}{\rho_v}} \quad \text{donne}$$

$$\frac{\Delta P_{TP}}{\Delta P_v} \quad \text{en fonction de} \quad \frac{\Delta P_v}{\Delta P_\ell} = \frac{\rho_L}{\rho_v} \times \left(\frac{\mu_v}{\mu_L}\right)^{0,25}$$

(Les ΔP_L ou ΔP_v étant calculés sur le débit massique total, sur toute la section de passage, avec les caractéristiques de la phase considérée).

Pour mémoire, citons aussi une méthode simple de calcul de ΔP_{TP} qui consiste à utiliser l'équation de Fanning où ρ est une densité apparente moyenne

$$\rho_M = \frac{W_T}{\frac{W_v}{\rho_v} + \frac{W_L}{\rho_L}}$$

Les pertes de charge étant déterminées, on calcule alors les cœfficients d'échange :

Si on utilise une méthode comme celle de Gloyer, on calcule les cœfficients d'échange en monophasique.

On connait :

- les vitesses de chaque phase
- leurs caractéristiques physiques
- la façon dont elles se répartissent (sur la paroi ou au centre du conduit)
  (méthode utilisée en condensation)

Si on utilise une méthode comme celle de Martinelli Nelson, on calcule un cœfficient diphasique

$$h_{TP} = a \times \left(\frac{1}{X_{TT}}\right)^{0.5} \times h_L + b\ h\ b$$

h_L est le cœfficient liquide supposé seul à circuler

h_b est un cœfficient fonction de la température, de la forme

$\alpha \, \Delta T^{\beta}_{\substack{\text{paroi} \\ \text{fluide}}}$ pour $\Delta T > 7°C$

et constant pour $\Delta T > 7°C$

a est un cœfficient constant

b est un cœfficient qui dépend du régime d'écoulement fonction de la vitesse massique et de X_{TT}, donné par abaque.

Cette méthode est surtout utilisée pour les échanges en vaporisation. Dans les échanges en vaporisation et en condensation de mélanges, où ces phénomènes se produisent à température variable, une part non négligeable de l'échange a trait aux chaleurs sensibles de la vapeur et du liquide, le reste étant affecté à une pseudo chaleur latente.

On effectue alors une pondération suivant ces chaleurs

- pour Gloyer de h_L par h_v.
- pour Martinelli Nelson de h_{TP} par h_v.

On peut utiliser également la méthode de Ward qui est du même genre, pour le calcul des échanges en condensation à l'extérieur des tubes d'échanges.

Les calculs, naturellement, sont en grande partie effectués sur ordinateur, ce qui permet d'obtenir des courbes de cœfficients d'échange et de pertes de charge fonction de la température donc de l'état du fluide.

Les différentes méthodes que l'on a pu évoquer ici ne peuvent pas tenir compte de tous les facteurs mis en jeu dans l'échange que nous avons à étudier. C'est pourquoi nous avons été amenés à établir un certain nombre de correlations originales qui nous ont permis de corriger ou de compléter les méthodes théoriques de calculs adoptées, afin de les amener en bonne concordance avec les résultats expérimentaux que nous avons obtenus.

C. DETERMINATION EXPERIMENTALE

Nous avons confronté nos méthodes de calcul avec des valeurs expérimentales déterminées de la façon suivante : d'une part, des mesures directes de cœfficients d'échange ont été effectuées sur un échangeur dans lequel on a vaporisé des mélanges d'hydrocarbures de compositions choisies en dégageant une certaine chaleur à l'intérieur des tubes grâce à des résistances électriques. Un grand nombre de sondes judicieusement placées permettaient de déterminer le diagramme d'échange et, par conséquent, l'écart moyen. Connaissant la quantité de chaleur échangée (par mesure de l'énergie électrique dissipée), on pouvait alors calculer le cœfficient d'échange en vaporisation. Cette expérimentation a principalement permis de mettre en évidence l'influence des différents paramètres intervenant dans la détermination de ce cœfficient. On a pu ainsi corriger les méthodes de calcul préconisées afin de tenir compte de façon plus conforme à l'expérience des corrélations mises en évidence.
D'autre part, un contrôle des valeurs obtenues par le calcul tant pour les cœfficients d'équilibre que pour les cœfficients d'échange a pu être réalisé, à partir de relevés effectués sur le pilote, de la manière suivante : on a sélectionné une marche stable pendant plusieurs heures pendant laquelle toutes les données caractéristiques du fonctionnement ont été relevées. En particulier : pressions, températures, compositions et débits de tous les fluides, y compris le gaz naturel, à l'entrée et à la sortie de chaque échangeur. On a alors, en utilisant nos données habituelles de calcul, établi des courbes de condensation en retenant :

- la composition du gaz de cycle à l'aspiration du compresseur
- les pressions relevées
- les températures relevées pour les pots de séparation

Le calcul a permis ainsi de déterminer les températures des fluides se vaporisant, à l'entrée et à la sortie des échangeurs et on a comparé ces températures déterminées par le calcul à celles qui avaient été relevées expérimentalement. Comme on peut le constater, les écarts restent faibles.

La liste des températures est la suivante :

Relevées (°C)	Calculées (°C)
+14	+13,5
-27	-26,5
-72	-71,5
-73	-74
-90	-91
-99	-99,5
-125	-127,5

Certains écarts ne sont pas négligeables. Cependant, ils ne sont pas dans le même sens et se compensent dans une large mesure de telle sorte que le recoupement obtenu peut être considéré comme satisfaisant.

A partir de ces résultats de calcul, et connaissant les surfaces réelles qu'ont les échangeurs du pilote, on a pu calculer le cœfficient d'échange global moyen de chacun des échangeurs. Ce cœfficient, comparé à celui que l'on obtient par un calcul purement théorique, met en évidence une petite sécurité implicite dans le calcul, sécurité qu'il convient d'ailleurs de conserver pour tenir compte de l'imprécision de ces déterminations, qui est du même ordre de grandeur.

IV. CONCEPTION D'ENSEMBLE

En ce qui concerne l'ensemble de l'usine, la première particularité du procédé à cascade incorporée est l'utilisation d'un compresseur de cycle unique. C'est un avantage considérable qui se traduit par une conception de l'usine et une exploitation plus simple, donc par une économie sensible.

Evidemment, la puissance de liquéfaction étant concentrée sur une seule machine, il en résulte que celle-ci, pour une capacité donnée de la ligne de liquéfaction, sera d'une puissance beaucoup plus importante que dans d'autres procédés. Ceci ne constitue cependant pas une limitation de la taille des lignes car on peut très bien concevoir éventuellement la mise en parallèle de deux compresseurs identiques, alimentant une seule ligne.

Ce problème de la capacité maximum des lignes est d'ailleurs beaucoup plus lié à la machine d'entraînement et aux échangeurs qu'aux compresseurs eux-mêmes.

Au contraire, en raison de sa plus grande puissance, la machine unique permet d'obtenir un meilleur rendement que ne le feraient par exemple trois machines développant au total une puissance équivalente.

1°) CHOIX DU COMPRESSEUR

Pour les plus petites capacités, on prévoira l'utilisation de compresseurs centrifuges. Par contre, pour des lignes plus importantes, on entre dans le domaine des compresseurs axiaux.

C'est ainsi, qu'au-dessus d'une capacité de un milliard de m3/an, on sera amené à choisir raisonnablement un compresseur axial, dans la plupart des cas. Par contre, aux alentours de cette capacité, le choix sera guidé par les considérations suivantes :

- Le compresseur axial permet d'obtenir un rendement supérieur. Ceci se traduit par une économie sur la consommation de fuel qui peut être appréciable, étant donné les puissances mises en jeu

- Le compresseur axial permet d'obtenir une solution plus économique en ce qui concerne les investissements totaux de l'usine

- Le compresseur axial existe déjà dans l'industrie de façon assez classique pour des puissances de l'ordre de 35 000 KW correspondant à une ligne de liquéfaction d'une capacité avoisinant 1 milliard de m3 par an

- Le compresseur centrifuge est, cependant, une solution plus classique et mieux connue

- Le compresseur centrifuge est plus "souple" et pose de moins délicats problèmes de régulation parce que sa zone d'utilisation est plus étendue. Il exige moins de précautions en ce qui concerne le pompage et le calcul des vibrations. Cependant, le principe même du cycle permet de maintenir le compresseur au même point de fonctionnement, moyennant des dispositions appropriées.

En définitive, la solution axiale est plus économique ; la solution centrifuge est, actuellement, moins délicate, tant pour la conception que pour l'exploitation.

2°) CHOIX DU MODE D'ENTRAINEMENT

Le choix du mode d'entraînement dépend également de façon très étroite de la taille de la ligne de liquéfaction considérée : on doit distinguer, en effet, les appareils destinés à liquéfier le gaz naturel pour le transport, qui traitent généralement une capacité dépassant

500 millions de m3/an et les appareils de "peak-shaving" qui permettent de stocker le gaz à l'état liquide et dont l'ordre de grandeur de puissance est 10 fois plus faible. Dans ce dernier cas, on pourra envisager, selon le contexte économique et technique du projet, un moteur électrique ou une turbine à gaz, entraînant un compresseur centrifuge de préférence (si on est dans la zone d'utilisation des compresseurs centrifuges). On peut également utiliser un motocompresseur : c'est une solution acceptable dans le cas où le compresseur centrifuge est exclu de toutes façons.

Pour une usine de liquéfaction de grande capacité, le choix se fera entre une turbine à gaz et une turbine à vapeur. Pour des tailles de lignes excédant un milliard de Nm3/an de G.N.L., la turbine à vapeur s'impose si on utilise un compresseur unique car la puissance mise en jeu dépasse la limite de puissance actuelle de la plupart des turbines à gaz industrielles qui se situe entre 30 000 et 40 000 kW.

Par contre, pour des lignes de capacité inférieures à un milliard de Nm3/an, ou pour des lignes de capacité supérieure, dans le cas où deux compresseurs sont utilisés, la comparaison est délicate :

- La turbine à vapeur est très bien connue et constitue une solution aux références nombreuses

- La mise en route est longue, mais ne pose pas de problème délicat

- Le rendement global de la centrale vapeur et de la turbine est supérieur à celui de la turbine à gaz, permettant une économie sur la consommation de fuel

- La puissance disponible sur l'arbre dépend relativement peu des conditions extérieures

A l'opposé, la turbine à gaz présente les inconvénients suivants :

- Les références industrielles sont peu nombreuses pour les puissances de l'ordre de 30 000 kW et la conception est moins classique

- La mise en route nécessite un moteur auxiliaire de démarrage dont la puissance peut être de l'ordre de 2 000 kW

- La consommation de fuel est plus élevée

- La turbine à gaz est très sensible aux conditions extérieures, en particulier à la température de l'air ambiant

C'est ainsi que la puissance fournie sur l'arbre d'une turbine à gaz diminue d'environ 1% quand la température de l'air ambiant augmente de 1°C. C'est un inconvénient majeur dans le cas où le climat est tel que les variations de température diurnes ou annuelles sont importantes. Il est, en effet, nécessaire, pour les variations saisonnières, de surdimensionner les turbines et les échangeurs pour leur permettre de limiter la perte de production due aux températures d'air les plus élevées. D'autre part, des variations diurnes importantes entraînent alors une conduite plus délicate de l'installation qu'il faudra constamment adapter aux conditions nouvelles.

Par contre, les investissements sont plus réduits par comparaison avec l'ensemble centrale vapeur-turbine à vapeur.

- La suppression de tous les auxiliaires dont dépend la centrale vapeur permet de réduire le nombre des pannes. Il s'ensuit un meilleur cœfficient d'exploitation qui compense partiellement la dépense supérieure en fuel, d'autant plus que celui-ci est bon marché, en général, dans une usine de liquéfaction

En définitive, le choix devra être fait en fonction des critères spécifiques de chaque projet, le choix de la turbine à gaz menant, dans la mesure où les conditions climatiques sont stables, à une solution plus économique en ce qui concerne les investissements et sensiblement équivalente en ce qui concerne le coût d'exploitation.

3°) TAILLE DES LIGNES

D'une façon générale, il est économique, sur le plan des investissements et du personnel de conduite, de limiter le nombre des lignes de liquéfaction. C'est pourquoi, en général, on cherche à réaliser des lignes de capacité maximum compatible avec les équipements choisis.

La gamme des plus grosses turbines à gaz réalisées actuellement est très limitée et se situe entre 30 000 kW et 40 000 kW. On sera donc amené à choisir parmi celles-ci une turbine standard qui permette de réaliser une ligne de liquéfaction de la capacité désirée.

Des critères économiques permettront alors de définir les autres équipements (échangeurs et compresseur) pour tirer le meilleur parti de la turbine ; on peut être aussi amené, dans certains cas, à utiliser, pour alimenter une seule ligne de liquéfaction, deux compresseurs en parallèle, chacun entraîné par une turbine développant la moitié de la puissance totale nécessaire.

Par contre, si on utilise une turbine à vapeur, la limite de capacité d'une ligne est alors fixée par la taille des échangeurs utilisés. En effet, au-delà d'un certain diamètre et d'un certain poids, les difficultés de transport sont considérablement accrues. La limite ainsi imposée se situe actuellement entre 1, 5 et 2 milliards de m3 par an pour une ligne qui réalise une simple liquéfaction du gaz naturel sans extraction de produits lourds.

4°) OPTIMISATION (fig. 5)

Quand on fait le calcul d'un appareil de liquéfaction, on a la faculté, dans une certaine mesure, de choisir les écarts de température sur les échangeurs, c'est-à-dire en définitive la surface d'échange totale utilisée à laquelle il correspond alors une certaine dépense d'énergie. De façon plus précise, la surface d'échange est, en première approximation, inversement proportionnelle à l'écart de température moyen, alors que la dépense d'énergie croît avec cet écart. Des études faites dans un certain nombre de cas ont permis d'établir la courbe représentant la variation de surface d'échange spécifique (m2 par Nm3/h de gaz liquéfié) en fonction de la dépense d'énergie spécifique (kW/Nm3/h de gaz liquéfié).

Cette courbe se traduit en termes économiques par la superposition des trois courbes suivantes, en fonction de l'énergie spécifique :

- coût de la surface d'échange totale
- coût de la puissance installée
- coût de l'énergie totale dépensée pendant la durée de la période d'amortissement choisie

Le minimum, en définitive, se révèle assez peu marqué et fait apparaître, en général, une dépense d'énergie se situant aux alentours de 0, 35 kW/Nm3/h.

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Dans certains cas, d'autres considérations peuvent amener à déplacer légèrement le point nominal obtenu en raison des limites de capacité imposées par certains équipements (turbines à gaz par exemple).

5°) STOCKAGES

Du fait du sous-refroidissement poussé du G. N. L., il est possible d'éviter tout flash lors de la détente au stockage. On prévoit même, grâce à un léger sous-refroidissement supplémentaire, de compenser les apports de chaleur de la ligne de transfert de G. N. L. de telle sorte qu'il n'en résulte aucune évaporation. On peut réduire ainsi le débit de gaz provenant du stockage au seul boil-off qu'on pourra recomprimer pour l'injecter dans le réseau fuel-gaz par exemple.

6°) EXTRACTIONS

Il est nécessaire d'extraire certains hydrocarbures lourds du gaz naturel afin d'en tirer les constituants du réfrigérant. Cette extraction se fait à partir d'une fraction liquide soutirée au cours de la condensation du gaz naturel. Dans certains cas, sur un gaz riche en hydrocarbures lourds, on peut procéder à une extraction plus importante et séparer des produits purs pour les commercialiser.

L'azote, souvent présent en quantité non négligeable dans le gaz naturel, doit être extrait avant le sous-refroidissement final du G. N. L. En effet, sa présence abaisse considérablement la température d'ébullition du gaz liquéfié, si bien qu'elle exige, pour éviter tout flash à la détente, de sous-refroidir le liquide jusqu'à une température beaucoup plus basse, ce qui est très coûteux en énergie. Pour cette raison, on extrait la plus grande partie de l'azote dans une colonne de distillation sous pression. La fraction extraite, pouvant par exemple contenir 30% d'azote, est alors utilisée comme fuel-gaz pour les besoins de l'usine. (fig. 6).

L'objet de cet exposé n'est pas de décrire les solutions proposées, mais de passer en revue les principaux problèmes qui se sont posés lors de la mise au point du procédé à cascade incorporée.

Le rôle des études théoriques a été essentiellement de permettre d'aborder les problèmes sous un angle scientifique. Mais, en définitive, il faut souligner l'importance de l'expérimentation qui, seule, permet de contrôler la validité de certaines hypothèses ou, mieux encore, d'établir de nouvelles corrélations que le raisonnement n'avait pas prévues. D'autre part, il est particulièrement important, dans le domaine industriel, de disposer d'informations sur les performances d'un appareil en fonctionnement puisqu'elles concernent, en définitive, l'objet de ces études. A ce titre, l'expérimentation du pilote de Nantes et d'unités de peak-shaving, a été capitale.

Les différentes recherches, études et essais ainsi que ces mises en route d'installations industrielles, permettent en fait de réaliser en toute sécurité la deuxième génération des installations de liquéfaction de gaz naturel et, en particulier, l'usine de Skikda qui liquéfiera 4, 5 milliards de m3/an après extraction partielle de C2 et de C3, en utilisant 3 compresseurs axiaux de 70 000 kW environ, entraînés par turbine à vapeur et une surface d'échange totale de 105 000 m2.

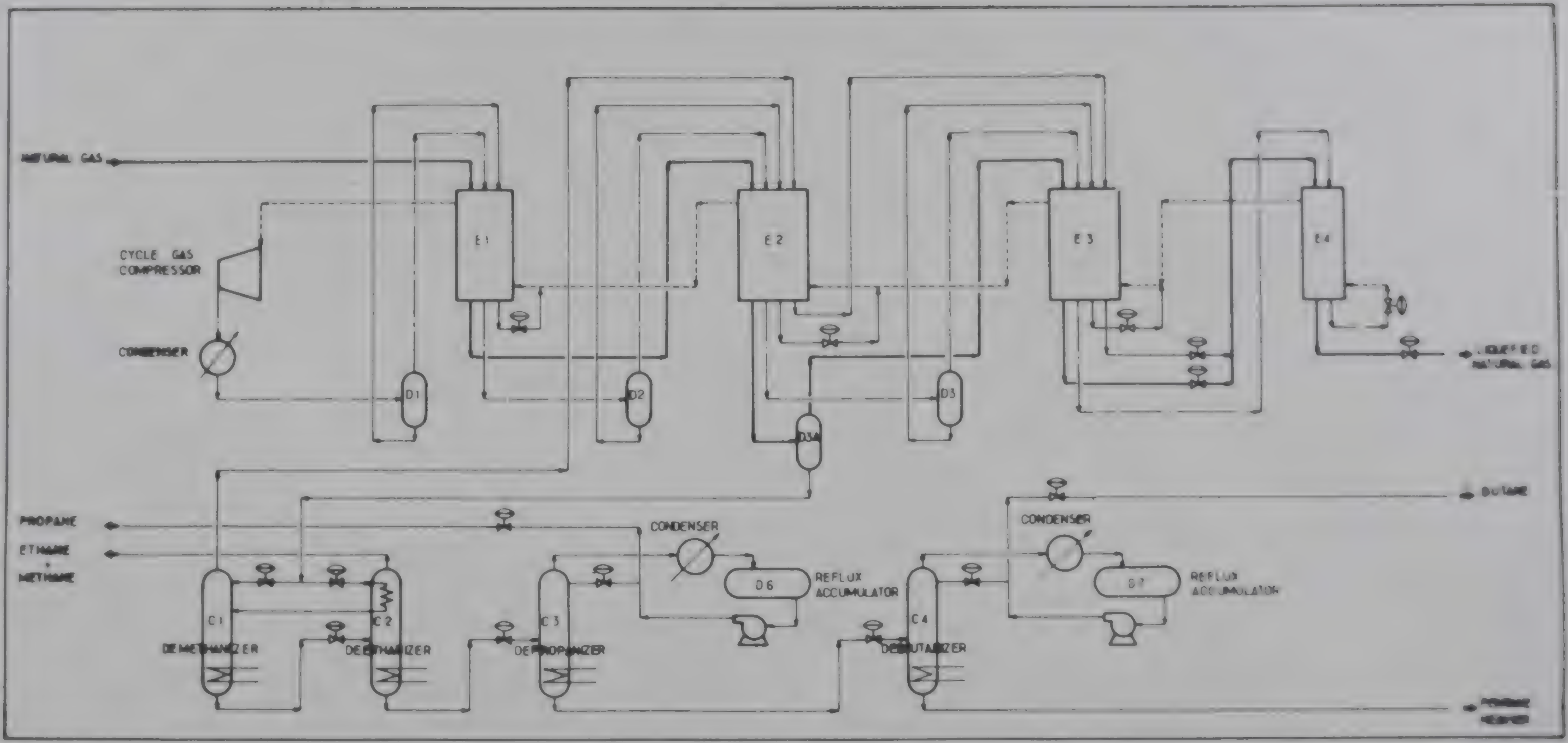

NATURAL GAS
CYCLE GAS COMPRESSOR
CONDENSER
E1
E2
E3
E4
D1
D2
D3
D3A
LIQUEFIED NATURAL GAS
BUTANE
PROPANE
ETHANE + METHANE
C1 DEMETHANIZER
C2 DEETHANIZER
C3 DEPROPANIZER
C4 DEBUTANIZER
CONDENSER
D6
REFLUX ACCUMULATOR
CONDENSER
D7
REFLUX ACCUMULATOR

Fig.1. Liquefaction de gaz naturel - cycle arc

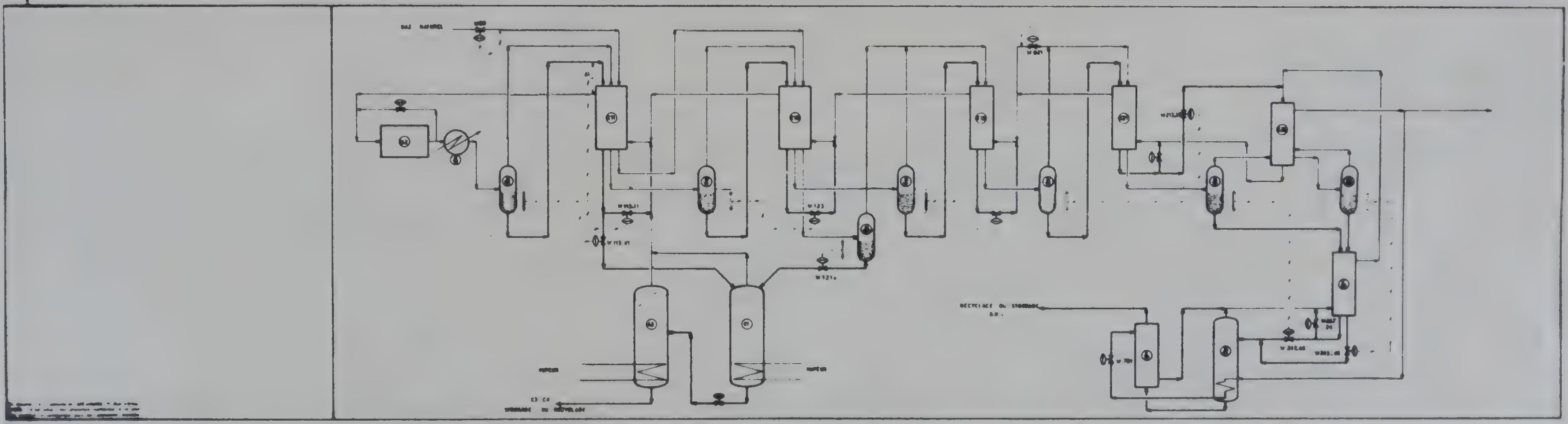

W 123
W 121a
C3 C4

Fig.2. Pilote de nantes

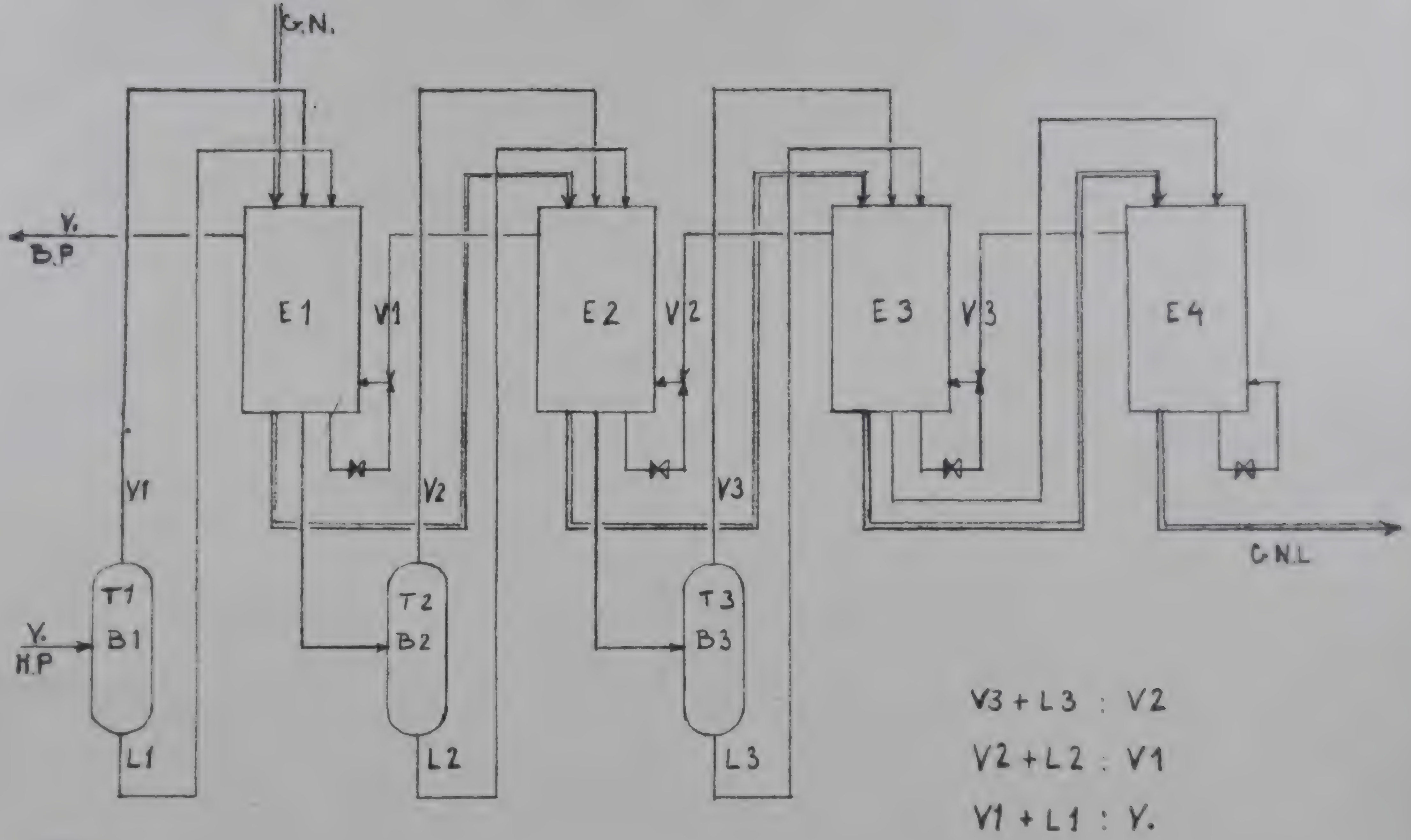
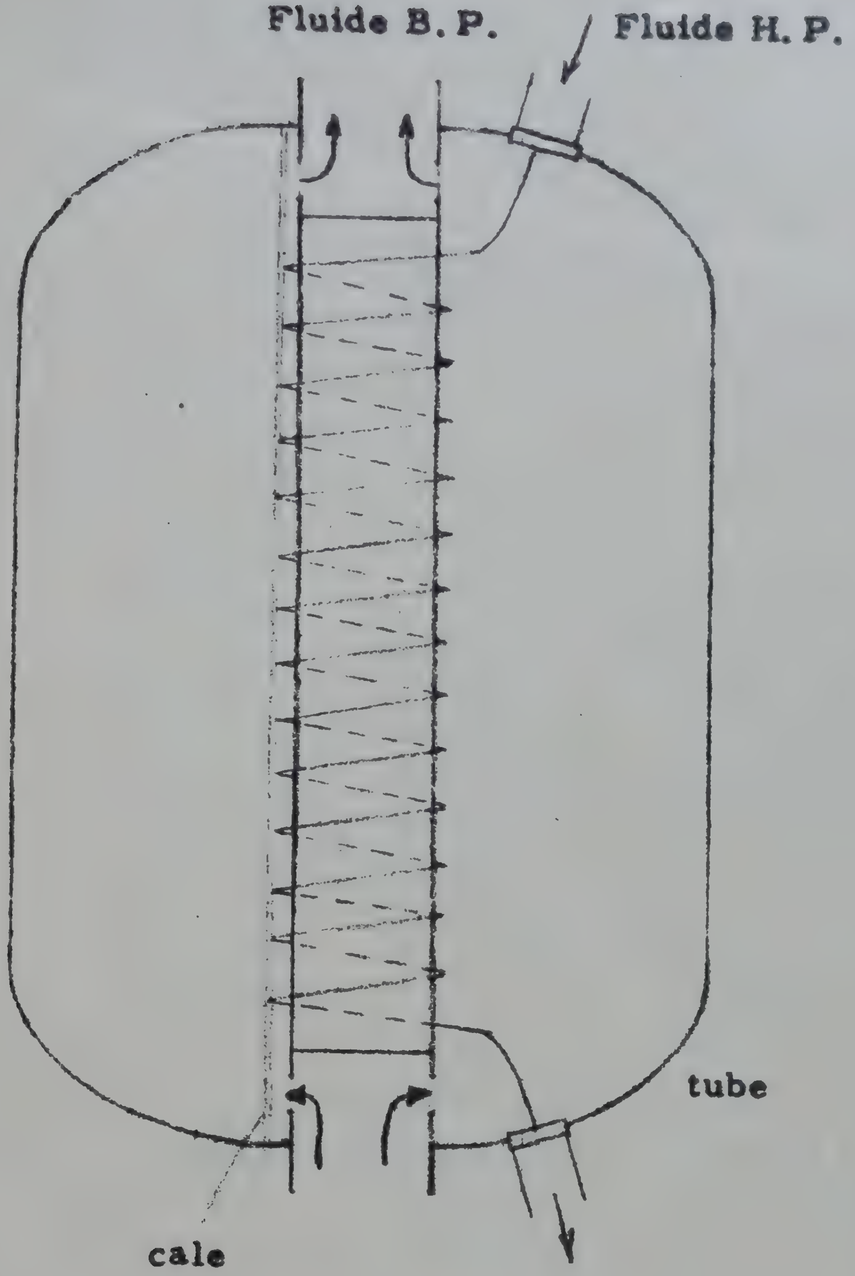
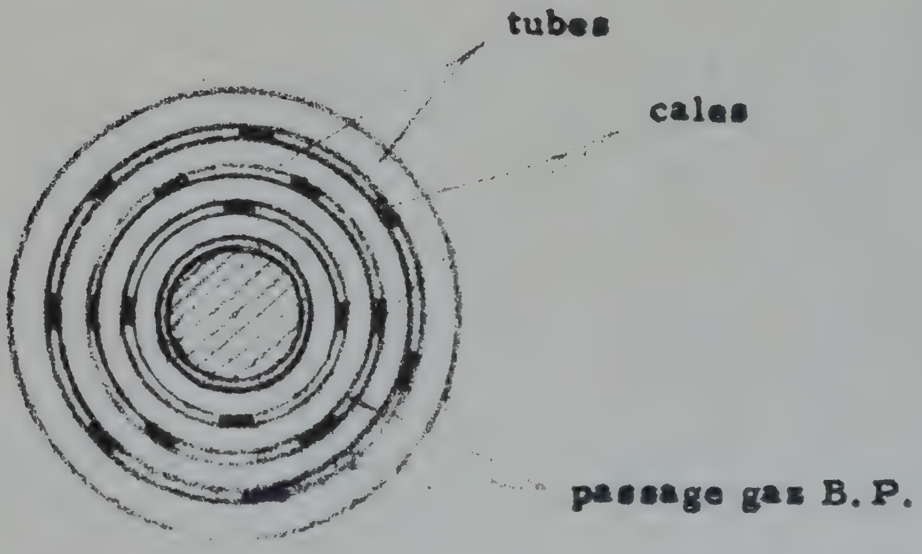

G.N.
Y. B.P
E 1
V1
E 2
V2
E 3
V3
E 4
V1
V2
V3
T1 B1
T2 B2
T3 B3
Y. H.P
L1
L2
L3
G.N.L
V3 + L3 : V2
V2 + L2 : V1
V1 + L1 : Y.

Fig.3. Cascade incorporee


Fluide B. P.
Fluide H. P.
tube
cale

Vue schématique


tubes
cales
passage gaz B. P.

Vue en coupe horizontale partielle

Fig.4. Echangeur bobine

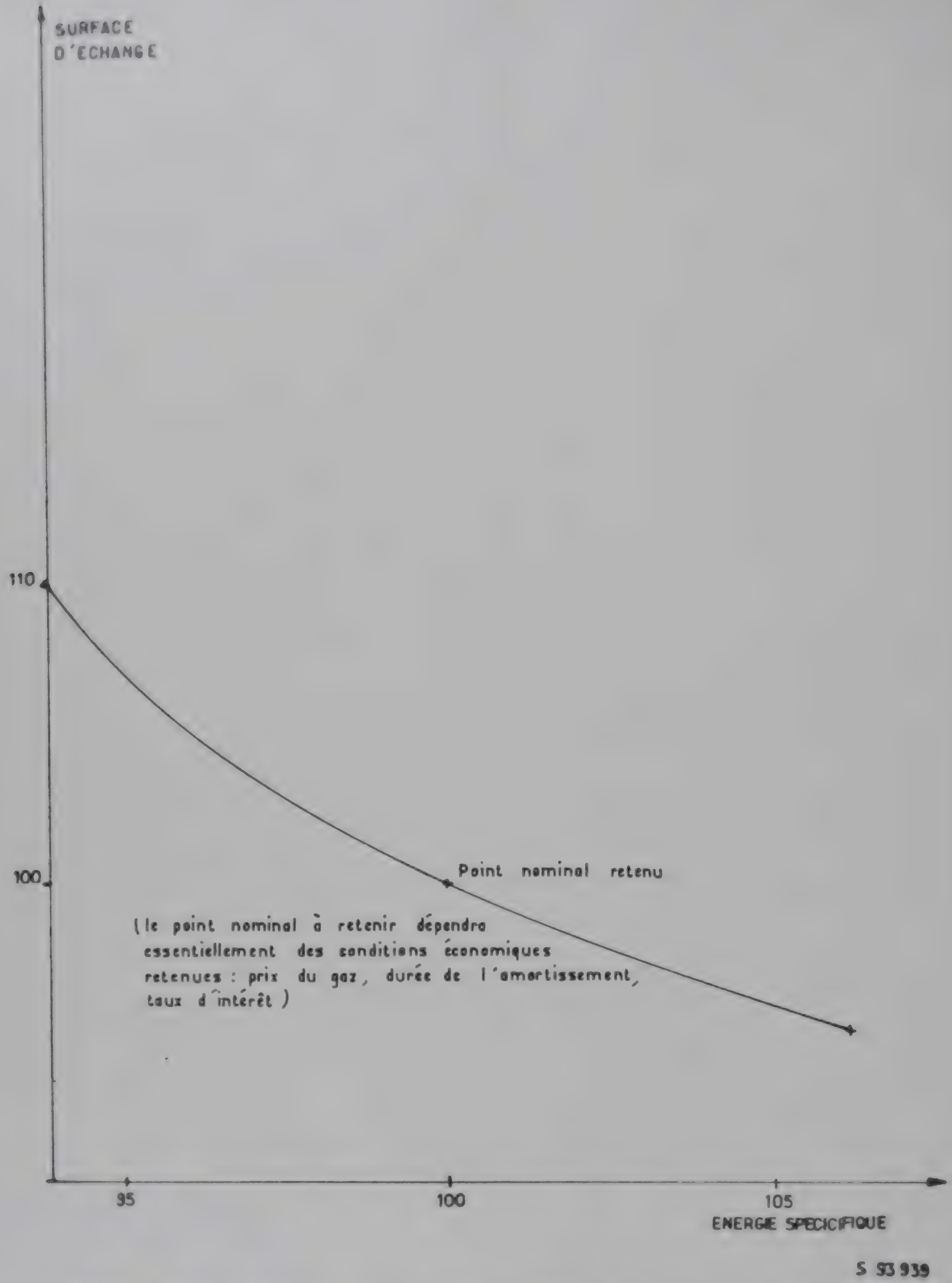
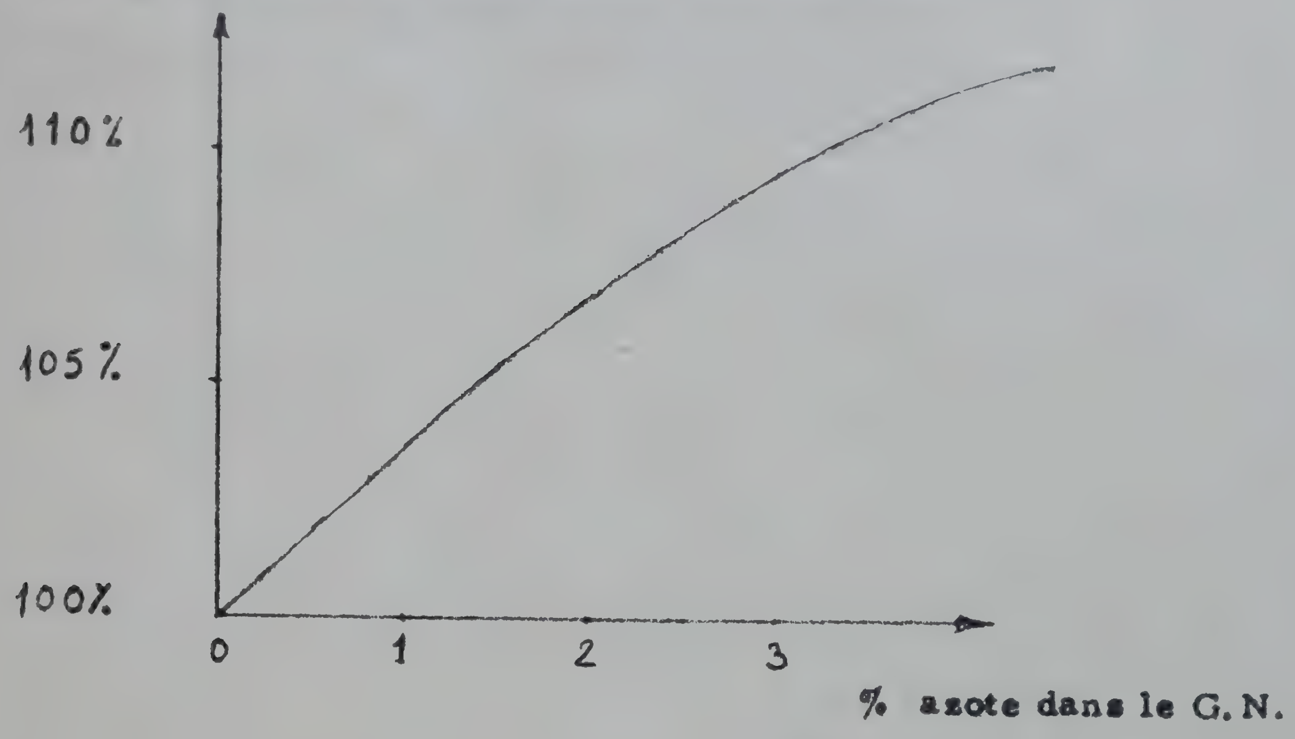
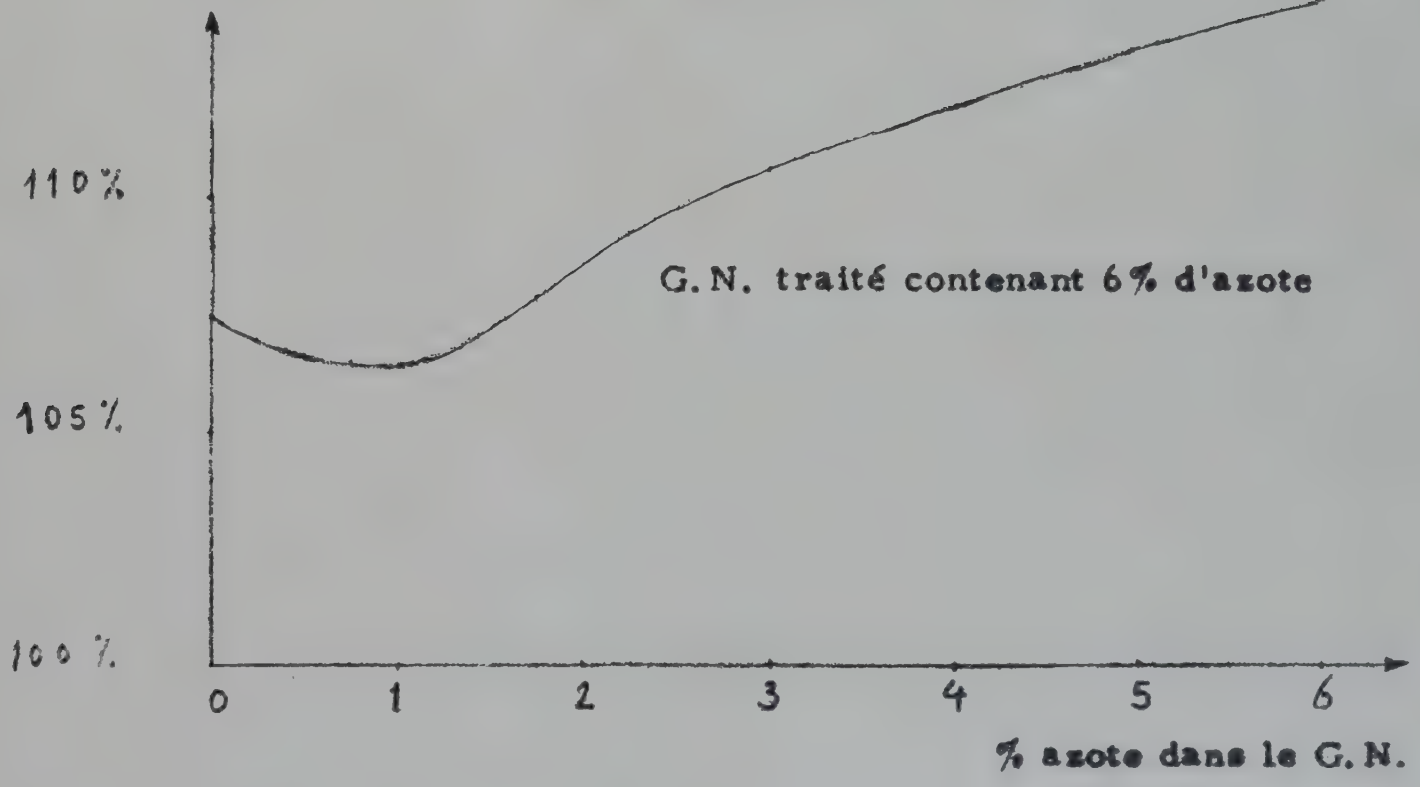


Fig.5. Cascade incorporee courbe depense d'energie en fonction de la surface d'echange



Energie spécifique (liquéfaction sans extraction)



Energie spécifique (liquéfaction avec extraction d'azote)

Fig.6.

THE PROCESS DESIGN AND OPTIMISATION OF A MIXED REFRIGERANT CASCADE PLANT

By D. T. Linnett and K. C. Smith*

1. INTRODUCTION

This paper discusses problems encountered in the design of Mixed Refrigerant Cascade (MRC) plants and the calculation methods developed and used by BOC-Airco to deal with these.

While many papers concerned with the application and operation of natural gas liquefaction circuits have been published, very little has been presented about the detailed process design of this type of plant. The design techniques available are, however, a crucial factor determining the capability of the contractor to produce an optimum solution for a given liquefaction duty. Where highly flexible calculation methods - such as those described here - are available, this will enable a comprehensive study of all relevant parameters to be made and thus facilitate both the selection of the best basic cycle and the optimum choice of process conditions within that cycle. Lack of sufficiently flexible methods will limit the number of variants which can be studied and the best answer may not be found.

These considerations are particularly relevant to MRC cycles in as much as the other two main alternative types of cycle employed for natural gas liquefaction, viz. the expander cycle and the "classic" cascade cycle, may be considered as being conventional from the design aspect, in that well established calculation procedures may be used. The MRC cycle, on the other hand, necessitates an entirely new approach to the development of flexible calculation methods.

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2.THE BASIC MRC CYCLE

The MRC cycle uses a single multicomponent refrigerant normally composed of constituents extracted from the natural gas undergoing liquefaction and normally using a single expansion ratio between the compressed refrigerant and all the return streams. The refrigerant cycle may be 'open' - in which the refrigerant and natural gas streams are combined and subsequently re-separated - or 'closed' - in which the natural gas stream is kept separate throughout. This paper deals specifically with 'closed' cycles, although the problems involved and the techniques developed are equally applicable to 'open'cycles.

A simplified flowsheet of a typical 'closed' MRC cycle is shown in Fig.1. The natural gas at supply pressure is cooled, liquefied and subcooled in five successive heat exchangers, the degree of subcooling being arranged to minimise flash on expansion to storage. At an intermediate point (12) a condensed 'heavies' fraction is separated from the natural gas and withdrawn from the liquefaction cycle. This fraction is separated into its constituents for use in making up refrigerant losses and adjusting the refrigerant composition.

The circulating refrigerant is compressed (1) and then subjected to stepwise condensation in a series of heat exchange stages at successively lower temperatures. After each condensation stage, the condensed fraction is separated from the uncondensed gas, sub-cooled in the next heat exchange stage and expanded through a valve. The expanded liquid joins with the return streams from the stages below and is evaporated and warmed to ambient temperature before being recycled to the refrigerant compressor (1).

It should be noted here that Fig.1 is merely indicative of a basic refrigeration cycle for an MRC plant. This will serve to illustrate the basic design problems involved.

3. THE BASIC DESIGN PROBLEMS

Referring to Fig.1 it can be seen that in this example the mixed refrigerant is separated into five fractions in successive stages of fractional condensation. These fractions comprise four liquids from separators 3,5,7 and 9 and the vapour from separator 9 which is subsequently liquefied in exchanger 10 and subcooled in exchanger 11. Each of the fractions provides the refrigeration requirements for the stage immediately following that in which it was condensed, e.g. the fraction of refrigerant condensed in 4 is separated in 5, subcooled in 6 and then evaporated at lower pressure in 6 to provide the refrigeration for that stage.

The quantity and composition of the fraction condensed in any stage must be arranged such that the refrigeration provided by its evaporation at the lower pressure will exactly match the requirements of the stage immediately following. In addition the composition of each liquid fraction must be such that the profile of its evaporation curve will provide a positive temperature difference between the cooling and warming streams along the whole length of the relevent exchanger. Thus, the evaporation of the various fractions must occur over the complete temperature range from LNG to ambient and the total refrigerant must contain sufficient components to provide the necessary degree of flexibility.

The various intermediate fractions are closely interrelated since they all derive from a single circulating fluid. Any change in the quantity and composition of liquid separated in one stage will affect the equilibrium in other stages. This will in turn affect the interstage temperatures and the temperature differences of the heat exchangers. In fact, virtually all the process variables are inter-dependent.

In order to illustrate the kind of problems involved in calculating such a cycle, let us consider a stage-wise calculation starting at the cold end of the cycle. Firstly, the composition and flowrate of the vapour leaving separator 9 and the interstage temperatures between exchangers 10 and 11 must be assumed before a trial and error heat balance can be carried out over exchanger 11. The liquid composition and quantity leaving separator 9 and the next set of interstage temperatures must then be assumed before a trial and error heat balance can be carried out over exchanger 10. Each trial for exchanger 10 reflects back on exchanger 11 resulting in a new trial and error calculation for that exchanger. Thus, as one progresses further along the cycle, the number of assumptions increases while each assumption involves new trial and error routines and each trial reflects back through the system to the starting point. The heavier hydrocarbons virtually disappear from the remaining refrigerant in the colder sections of the plant and must therefore be introduced into the calculations as they become significant. At each stage, the operability of the heat exchangers must be checked and the interstage temperatures and temperature differences must be optimised. Moreover, since the optimum solution is an economic balance between power consumption and overall capital cost, the apparent optimum solution for any one stage does not necessarily correspond to an optimum solution overall. Indeed, any improvement in the cycle efficiency at one point may well impose worse or inoperable conditions at another point in the cycle.

Such a step-wise calculation procedure thus involves an extremely complex nest of trial and error calculations with too little flexibility to permit proper optimisation of the overall process. A similar situation arises if one starts at any other point in the cycle and we concluded that a completely different approach should be adopted.

Any alternative calculation procedure must still be highly complex and the use of a computer is essential for effective analysis. This leads to the requirement for all the necessary basic data to be in a form suitable for computer calculations. The mixed refrigerant may contain five or more components and consistent equilibrium and enthalpy data are required over very wide ranges of composition, pressure and temperature. In addition, density, thermal conductivity and viscosity data for multicomponent gas and liquid mixtures are required for heat exchanger sizing.

4. POSSIBLE VARIATIONS AND ADDITIONS TO THE BASIC CYCLE

The foregoing discussion has been related merely to the basic refrigeration cycle. Design methods will have to cater for more complicated arrangements than this in order to produce optimum solutions for particular plant specifications. Some of the possible additional requirements are given below.

(a) Number of Heat Exchanger Stages

The object of the successive condensation stages is to match refrigeration production as closely as possible to the requirements at any temperature level. The closer the match achieved, the higher the thermodynamic efficiency of the cycle and the lower its power consumption. A practical limit is set by the number and size of heat exchangers which are economically acceptable, consistent with the lowest overall operating cost. The degree of optimisation possible is dependant upon the flexibility of the stage compositions which is in turn dependant upon the number of components present in the refrigerant. Thus the number of heat exchange stages will depend on the make-up of the refrigerant, the size of plant and the relative importance of capital cost versus power.

(b) Arrangements of Heat Exchangers

Fig.1 shows a single heat exchanger in each refrigeration stage. In practise, a combination of heat exchangers may have to be considered for any of a number of reasons e.g. improved efficiency, control purposes, inclusion of additional side streams etc.

(c) Storage Tank Boil-Off

Re-liquefaction of the storage tank boil-off is frequently necessary in order to maintain the composition of the stored liquid. Alternatively it may only be necessary to recover refrigeration from the boil-off before discharge into a gas distribution line.

(d) Separation of Nitrogen

Nitrogen has no calorific value and, depending on its concentration in the natural gas, may have to be removed. Alternatively, a nitrogen fraction may be separated for subsequent recovery of helium should this be required. In either case a degasification column would be incorporated with the added complication of providing reboil and reflux at the appropriate temperature levels.

In addition to the main cycle variations, other factors must be taken into account during process design and optimisation for a particular plant duty. Examples of these are:

(a) The mixed refrigerant will normally comprise components extracted from the natural gas in order to avoid the use of imported refrigerants. The range of components comprising the refrigerant stream will therefore depend partly on the natural gas composition and partly on such factors as the evaporation characteristics of the available components and the economical number of refrigeration stages.

(b) The plant may have to be designed for a range of natural gas composition, pressures and temperatures.

(c) By-products may be required, in which case additional duties may be imposed on the circulating refrigerant.

(d) Plant control must be considered in respect of detailed cycle selection and control ranges and responses.

(e) Full account must be taken of any limitations imposed on the operating conditions due to equipment selection or mechanical design considerations, e.g.

    (i) rotary or reciprocating compressors.
    (ii) gas engine, gas turbine or electric drive.
    (iii) type of heat exchanger to be used.
    (iv) type of pre-purification equipment and degree of purification required.

(v) limitations on size of cold box 'packages' for transport purposes.

5. DEVELOPMENT OF BASIC DATA

For any cycle calculation program reliable equilibrium and enthalpy data must be provided for the full composition, pressure and temperature ranges likely to be encountered. The choice is either to prepare the necessary data in a suitably correlated form such that it can be conveniently stored as tables/equations or to program a data prediction method which can calculate data as required. For more than three components the storage of composition dependant data in tabular form is impracticable. Data correlated in terms of convergence pressure, molal average boiling point or any other mixture parameter can be used but their accuracy and consistency will be limited by the number of tables which can be stored and the accuracy of interpolation.

For these reasons we decided to concentrate on generalised data prediction methods and, after surveying possible alternatives, selected the Benedict-Webb-Rubin equation of state as being most likely to fulfil our requirements. Our development programme on this method included testing various sets of published constants and calculation of alternative sets of constants from experimental PVT data. Refinements of the basic calculation procedure to provide the best fit to experimental equilibrium and enthalpy data e.g. C_o and γ variation, were introduced as appropriate. Initially we aimed at reproducing enthalpy and vapour pressure data for each of the pure components. The work was later extended to equilibria and enthalpies of mixtures with particular attention being paid to the lower temperature regions.

Once we were satisfied with the consistency and accuracy of the prediction of both enthalpy and equilibrium data for nitrogen - light hydrocarbon mixtures, data sub-routines were prepared which could be called by any main calculation to return necessary data. These routines were used in the development of design methods for MRC plants and are now in general use within the Company in other cycle calculation programs. In their current form they will return equilibrium data together with enthalpies, entropies and densities of gas, liquid or two phase mixtures as required.

Using the BWR sub-routines, further sub-routines were written for calculation of bubble points, dew points, exchanger $Q/\Delta T$'s, temperature from enthalpy, flash on expansion and other functions which may be used repetitively in the course of cycle calculations. Additional computer programs were prepared for calculation of viscosities, thermal conductivities and specific heats of gas, liquid and two-phase mixtures, thus fulfilling all our basic data requirements.

6. DEVELOPMENT OF DESIGN METHODS

Referring again to Fig.1, the main process variables for a specified plant duty are:-

(a) Stage temperatures : T_1, T_4, T_7, T_{10}, T_{13}, T_{16}.

(b) Exchanger W.E.T.D's T_1-T_3, T_4-T_6, T_7-T_9, T_{10}-T_{12}, T_{13}-T_{15}.

(c) Return stream temperatures : T_5, T_8, T_{11}, T_{14}, T_{17}.

(d) Refrigerant pressures : P_1, P_2.

(e) Overall refrigerant composition and flowrate.

(f) Composition and flowrate of each refrigerant stream.

Bearing in mind that the refrigerant may contain five or six components and the composition of any fraction therefore represents several variables, it is clear that we have a large number of inter-related effects to consider. Having discarded step-wise calculation procedures because of their complexity (each process variable represents a trial and error calculation) and limited flexibility, we concentrated on developing a method of calculating all parts of the cycle simultaneously.

Qualitative analysis of the process shows that certain of the variables are directly related, e.g. the compositions and flows of the refrigerant fractions directly affect each other and their respective stage temperatures. Other variables are only indirectly related, e.g. T_4 and T_{10} are related only through the inter-dependence of the refrigerant fractions. After studying the various relationships in this way, we selected certain of the variables as the most suitable to be considered as "fixed" quantities. All the remaining variables were treated as 'floating' quantities. We then built up a complete cycle calculation which incorporated all the 'floating' variables in terms of a number of simultaneous equations. The equations form the basis of a single iterative loop by which the values of all the 'floating' variables converge simultaneously to give a completely balanced solution to the whole cycle. To complete the optimisation of cycle operating conditions, ranges of values are investigated for each of the 'fixed' variables.

Throughout the calculation, equilibrium and enthalpy data are provided by the BWR sub-routines as required. After completion of the convergence calculations, exchanger operabilities are checked and $Q/\Delta T$'s calculated. The output from the program includes all process details necessary for equipment design and the converged values for the process variables listed above.

The manner in which the equations and convergence procedures are set up and programmed provides the necessary high degree of flexibility in the calculations. Any combination of the cycle variations and additions described in section 4 can be adequately handled. These affect only the form of the equation coefficients and do not affect the calculation procedure itself. Full account is also taken of such factors as heat inleak and stream pressure drops.

With the completed program, process design and full optimisation can be carried out for an MRC plant tailor-made to fit particular duty specifications. In addition, control ranges and responses can be investigated and the effects of changes in the plant operating conditions can be determined.

7. TYPICAL RESULTS FROM THE CYCLE CALCULATION PROGRAM

As previously discussed, there are many possible alternative forms of the MRC cycle, the preferred form depending on the specific requirements for any particular plant. As an illustration of possible operating conditions, some typical results for two relatively straight-forward specific cases are given below. Both cases are for liquefaction of North Sea type gas with a product rate of 400 long tons/day net (480 T/D gross) into storage at atmospheric pressure. The degree of sub-cooling of the LNG is arranged so that no flash occurs on expansion into storage. The storage tank boil-off fraction is not re-liquefied but its refrigeration is recovered in the heat exchange system. In both cases the refrigerant stream consists of nitrogen, methane, ethane, propane, normal-butane and iso-butane.

Fig.2 shows a liquefaction cycle which is essentially similar to the basic cycle in Fig.1, but includes the boil-off and refrigerant make-up streams.

Fig.3 shows a liquefaction cycle incorporating the same features as Fig.2 but with the additional separation of a nitrogen rich fraction in a degasification column. In addition the third stage heat exchanger is split to provide more flexible control of the column operating conditions.

Design Basis

Natural gas composition after pre-purification		% by volume
	N_2	2.31
	CH_4	93.04
	C_2H_6	3.63
	C_3H_8	0.77
	n-C_4H_{10}	0.13
	i-C_4H_{10}	0.12

Natural gas pressure (psia) 588

LNG product flowrate (Sm^3/hr) 28,550

Calculated Refrigerant Details		Cycle in Fig.2	Cycle in Fig.3
Refrigerant composition (% by volume)	N_2	10.44	10.05
	CH_4	37.26	45.95
	C_2H_6	31.82	22.72
	C_3H_8	3.50	5.00
	n-C_4H_{10}	9.20	8.81
	i-C_4H_{10}	7.78	7.47

Refrigerant flowrate (Sm^3/hr)	107,100	120,070
Refrigerant pressure : high (psia)	588	588
low (psia)	80	80
Refrigerant compressor power at shaft (kW) :	9,460	10,600

The above conditions for the cycle in Fig.3 are for total removal of nitrogen from the natural gas, the nitrogen being concentrated to 30% in the column overhead stream. The resultant increase in power reflects the work required to achieve this separation.

8. POSSIBLE WAYS OF INCREASING CYCLE EFFICIENCY

The lowest practical power consumption for a particular cycle will be achieved by obtaining the best possible match of the enthalpy-temperature profiles in the heat exchangers consistent with the lowest overall operating cost. The cycle efficiency can be increased in a number of ways, e.g. increase in the number of heat exchange stages or use of multiple heat exchangers within any stage. Such methods will result in a significant increase in capital cost, however, and may not be justifiable in terms of overall operating cost.

An alternative method of improving efficiency, for which a patent application has been made, is to increase the flexibility in the compositions of the various refrigerant fractions. This can be achieved either by intermixing controlled quantities of different refrigerant fractions or by injection of controlled quantities of pure components into any of these fractions. By this means the composition of any fraction can be adjusted such that its evaporation curve more closely matches the requirements for the relevant heat exchange stage.

The essentials of this method when applied to the cycle in Fig.2 are shown in Fig.4. Streams A_1, A_2, A_3 represent a coarse adjustment in which a controlled quantity of liquid condensed in any stage is added to the liquid condensed in the following stage. Streams B_1, B_2 represent the fine adjustment of the composition of liquid to be evaporated in exchanger 8 by addition of one or more pure components. In order to maintain the overall composition of the circulating refrigerant, a bleed is taken from one or more of the condensed liquids, e.g. streams D_1, D_2 and fed to the column separating systems. The injection rates and the withdrawal rates of each component are balanced accordingly.

As an example of the increase in efficiency which can be obtained by this method, our calculations show a 3% reduction in power for this particular cycle by the optimum use of stream A_3 alone. The increase in capital cost associated with this power reduction is negligible.

9. OPTIMISATION OF HEAT EXCHANGER SYSTEM

Fig.5 shows a typical set of combined enthalpy/temperature profiles for the heat exchanger system illustrated in Fig.2. The lower line represents the natural gas cooling curve combined with the condensing and subcooling curves for the various refrigerant fractions. The upper line represents the combined curve for the return streams. The bulk of the natural gas condenses in exchanger 8, although the effect of this latent heat region on the combined H/T curve is partially masked by the continuous condensation of refrigerant throughout exchangers 4, 6, 8 and 10. The isenthalpic drop in temperature of the return streams at the cold ends of exchangers

4, 8 and 10 result from the Joule-Thompson cooling on expansion of the respective sub-cooled liquid fractions. No such effect is obtained at the cold end of exchanger 6 since in this case the degree of subcooling is such that no flash occurs on expansion.

As mentioned previously, there must be a positive temperature difference throughout the heat exchange cycle and the power consumption is minimised by making the temperature differences as small as possible. The curves shown represent the overall optimum solution for the particular case considered. For example, closing up the cold end temperature differences in exchangers 4 and 8 results in wider temperature differences at both ends of exchanger 6 for operable conditions and the overall power consumption increases. This is due to the inter-dependence of all the process variables and illustrates the necessity of obtaining an overall optimum solution rather than attempting to optimise each exchanger. It is possible to obtain a better match in exchanger 8 by lowering its warm end temperature to coincide with the bulk condensation of natural gas. However, this would necessitate additional complication of heat exchangers and controls to obtain the required temperature for separation of a suitable natural gas fraction in item 12.

Additional optimisation of individual heat exchangers can be effected by the method described in section 8. The broken line on Fig.5 shows the effect of using stream A3 in Fig.4. For this particular case, only a small improvement can be obtained with streams A_1 and A_2 since they partially affect the already fully optimised exchanger 6. Further optimisation of exchangers 4, 8 and 10 can be obtained by the addition of one or more pure components to the respective refrigerant fractions.

Streams A_1 and A_2 may be used effectively in other applications where different H/T profiles may result in a completely different overall optimum solution.

It would be extremely difficult to cater for the intermixing of stage liquids and addition of pure components to individual fractions in any step-wise calculation procedure. Because of the simultaneous convergence techniques used in our design program, however, it was possible to incorporate the necessary modifications to the program in a few hours. This illustrates the inherent flexibility of such calculation techniques.

ACKNOWLEDGEMENT

The authors are indebted to the Directors of BOC-Airco Cryogenic Plant Ltd for their kind permission to publish this paper.

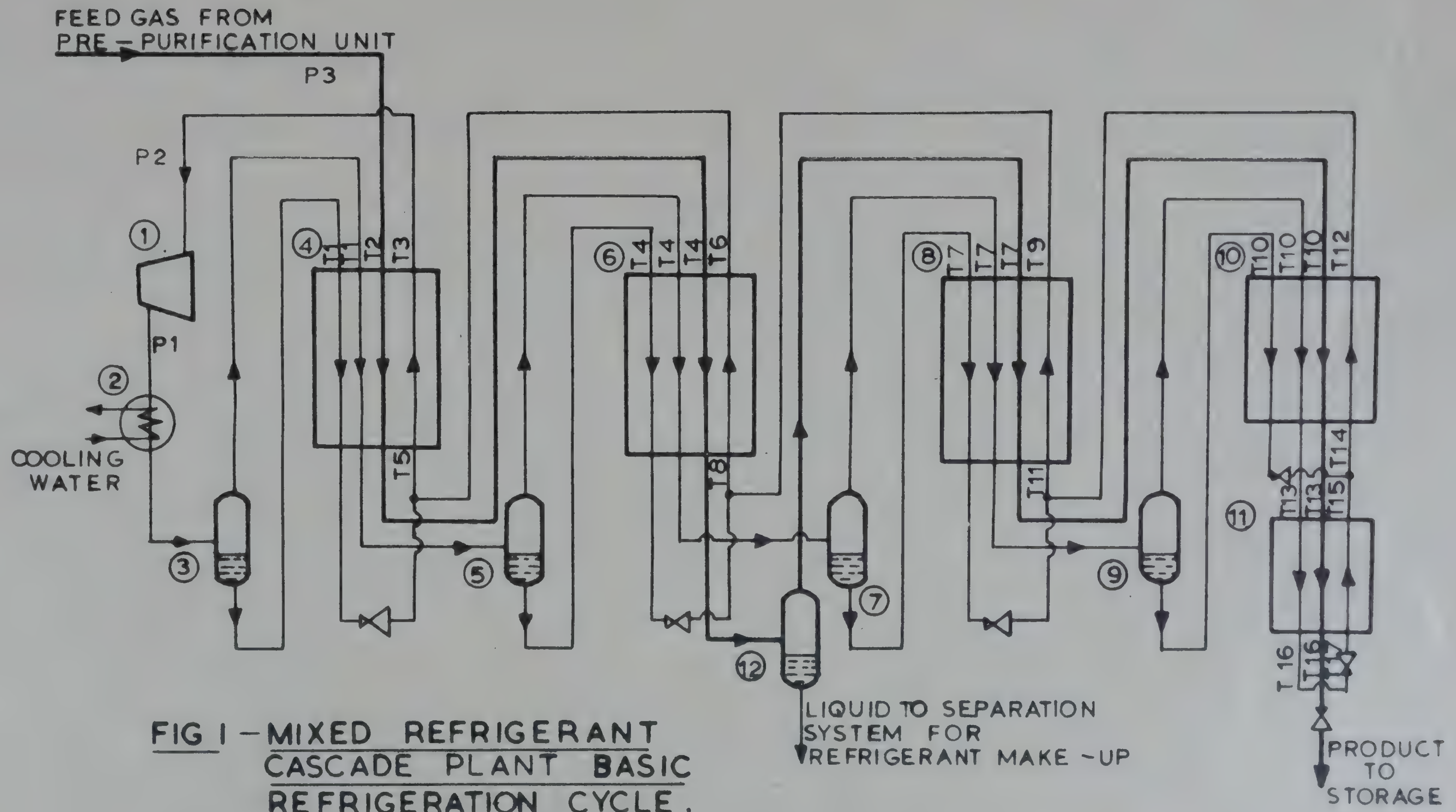

FEED GAS FROM
PRE - PURIFICATION UNIT
P3
P2
P1
①
②
COOLING
WATER
③
④
T1
T2
T3
T5
⑤
⑥
T4
T4
T4
T6
T8
⑦
⑧
T7
T7
T7
T9
T11
⑨
⑩
T10
T10
T10
T12
T14
T13
T13
T15
⑪
T 16
T16
⑫
LIQUID TO SEPARATION
SYSTEM FOR
REFRIGERANT MAKE - UP
PRODUCT
TO
STORAGE

FIG I - MIXED REFRIGERANT CASCADE PLANT BASIC REFRIGERATION CYCLE.

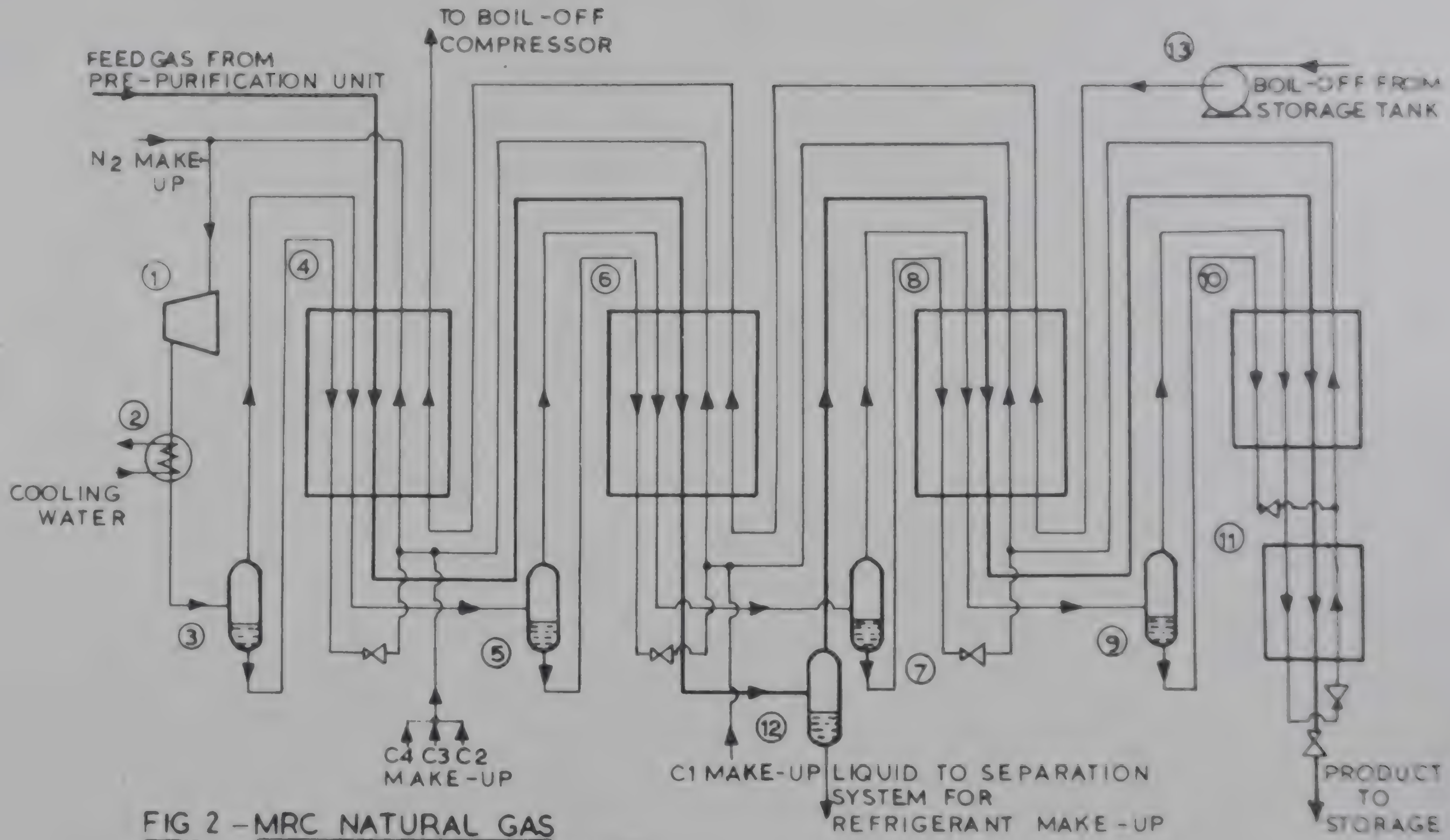

TO BOIL-OFF COMPRESSOR
FEEDGAS FROM PRE-PURIFICATION UNIT
N2 MAKE-UP
BOIL-OFF FROM STORAGE TANK
COOLING WATER
C4 C3 C2 MAKE-UP
C1 MAKE-UP
LIQUID TO SEPARATION SYSTEM FOR REFRIGERANT MAKE-UP
PRODUCT TO STORAGE
①
②
③
④
⑤
⑥
⑦
⑧
⑨
⑩
⑪
⑫
⑬

FIG 2 – MRC NATURAL GAS LIQUEFACTION PLANT (WITHOUT NITROGEN SEPARATION.)

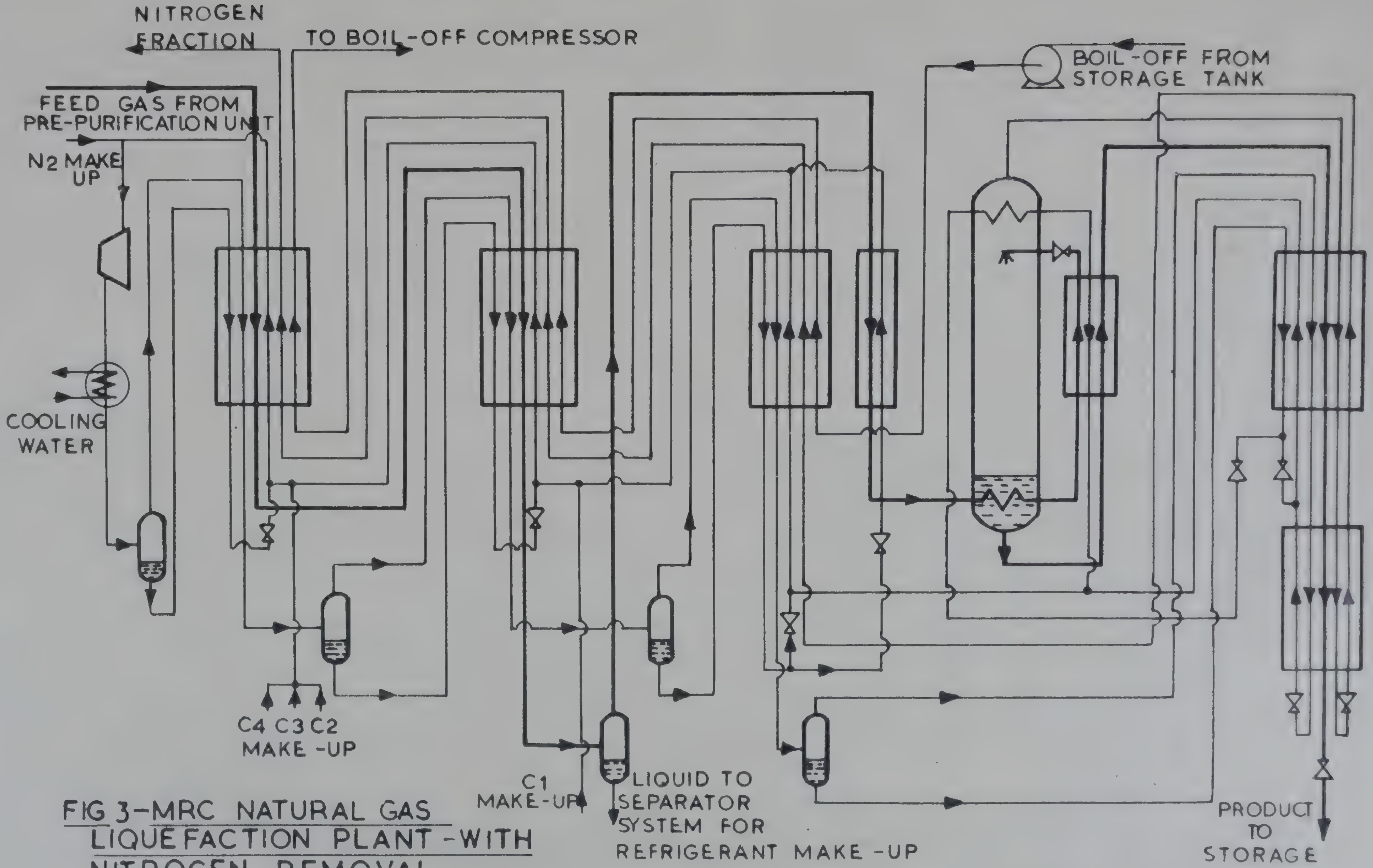

NITROGEN FRACTION
TO BOIL-OFF COMPRESSOR
FEED GAS FROM PRE-PURIFICATION UNIT
N2 MAKE UP
COOLING WATER
C4 C3 C2 MAKE -UP
C1 MAKE-UP
LIQUID TO SEPARATOR SYSTEM FOR REFRIGERANT MAKE -UP
BOIL-OFF FROM STORAGE TANK
PRODUCT TO STORAGE

FIG 3–MRC NATURAL GAS LIQUEFACTION PLANT - WITH NITROGEN REMOVAL

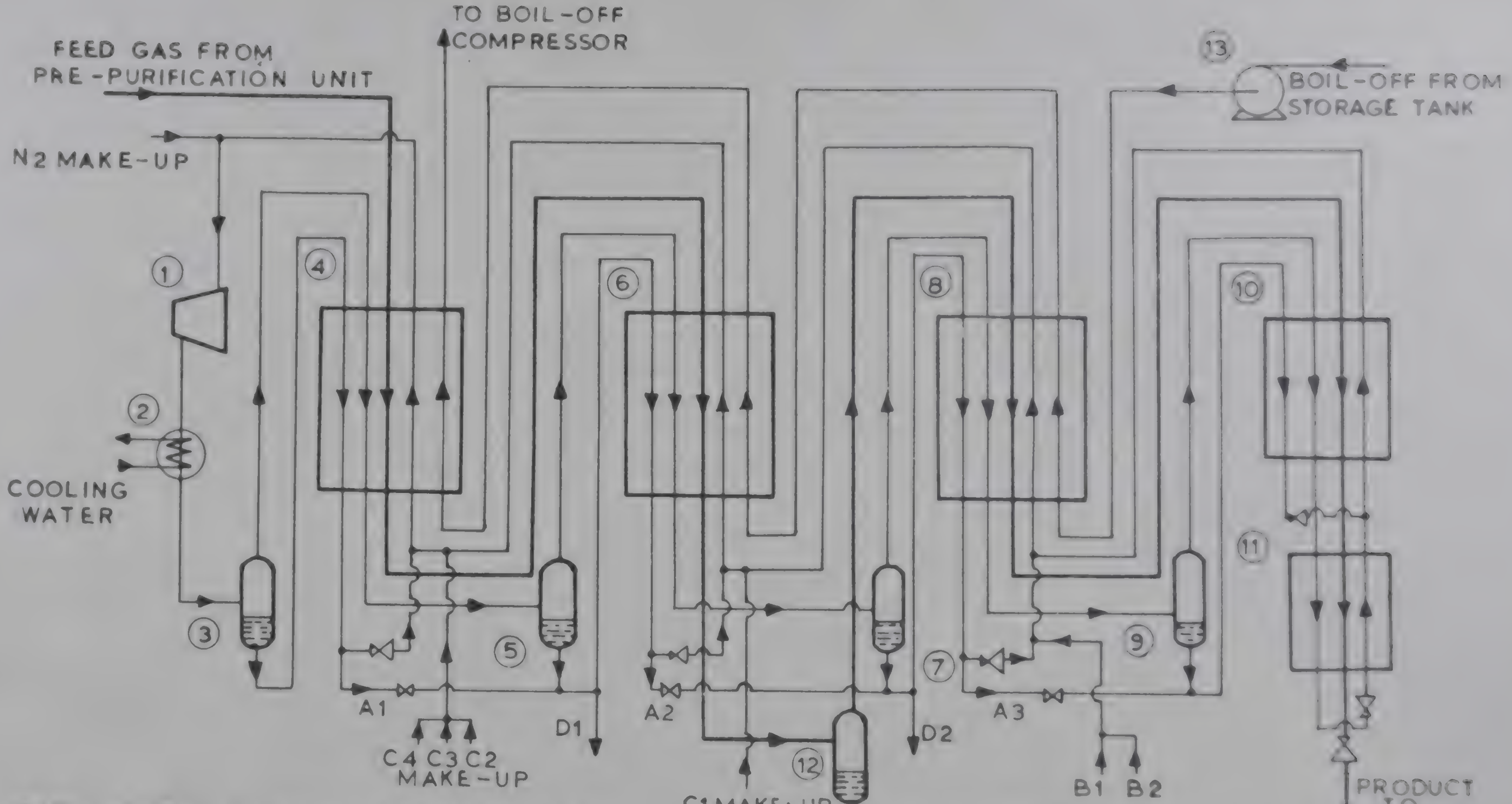
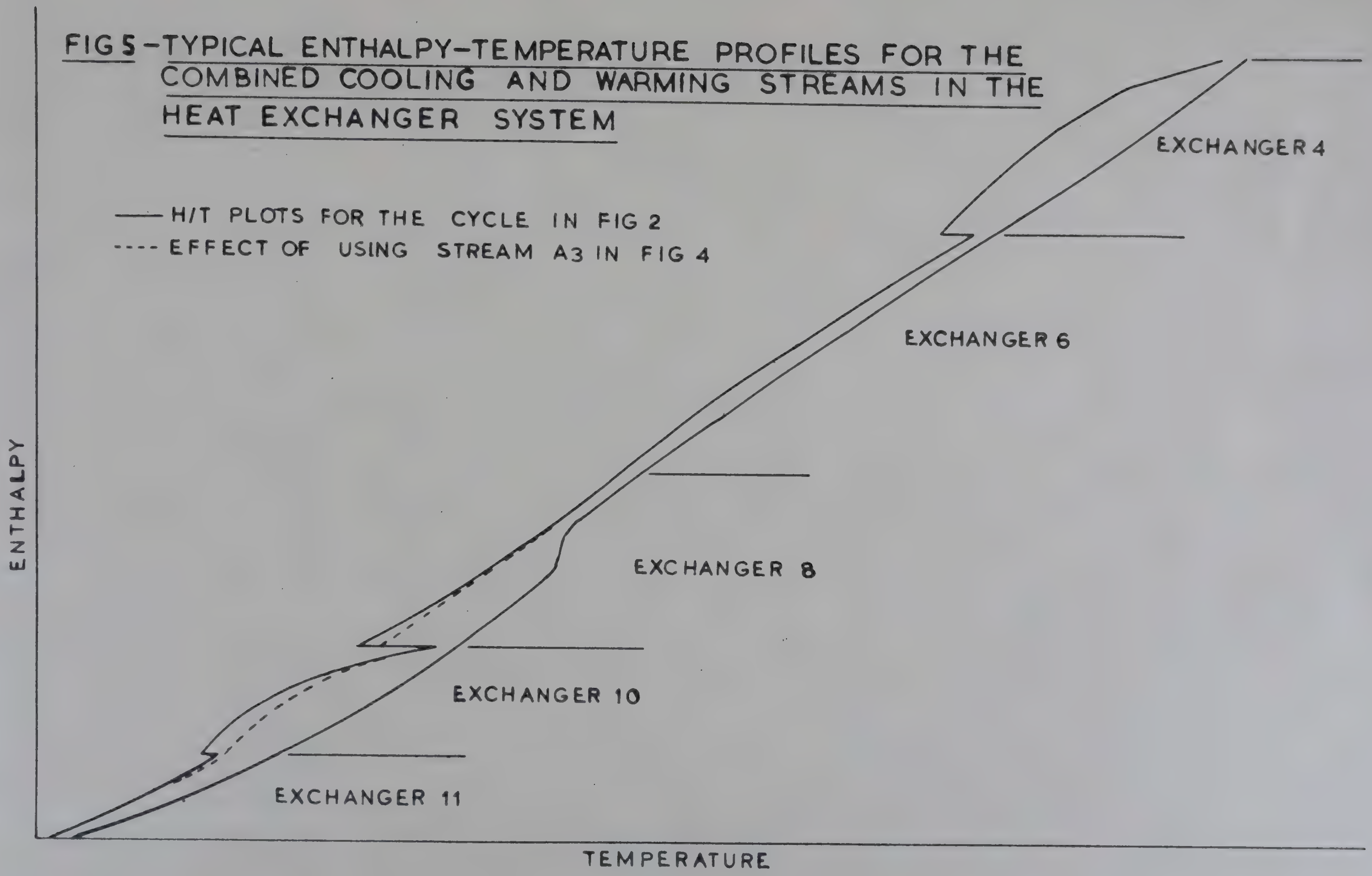

FEED GAS FROM PRE-PURIFICATION UNIT
N2 MAKE-UP
TO BOIL-OFF COMPRESSOR
BOIL-OFF FROM STORAGE TANK
COOLING WATER
①
②
③
④
⑤
⑥
⑦
⑧
⑨
⑩
⑪
⑫
⑬
A1
A2
A3
D1
D2
C4 C3 C2 MAKE-UP
C1 MAKE-UP
B1 B2
LIQUID TO SEPARATION SYSTEM FOR REFRIGERANT MAKE-UP
PRODUCT TO STORAGE

FIG 4 - MRC PLANT - METHOD FOR INCREASING CYCLE EFFICIENCY

FIG 5-TYPICAL ENTHALPY-TEMPERATURE PROFILES FOR THE COMBINED COOLING AND WARMING STREAMS IN THE HEAT EXCHANGER SYSTEM



THE INFLUENCE OF 'REAL-GAS' BEHAVIOUR IN MODERN COMPRESSOR DESIGN

By F. Weal

A) To fulfill guaranteed compressor performance in the pressure-, volume- und temperature characteristics, in the power consumption and in the cooler heat transfer, the deviation of the medium from the "ideal gas" must strictly be considered, whenever the kind of gas, resp. the pressure and temperature range to be covered warrant it. Knowledge of the behaviour of gases within the compressor cycle must go parallel with research work on compressor elements to allow a design with very close tolerance. Furthermore the correct interpretation of compressor test results, accurate flow measurements, usage of laws of similitude and other features necessitates such knowledge.

B) Some aspects of the behaviour of real gases will be discussed with the help of a new diagram based on the law of "corresponding states": $p_r \, v'_r = z \, . \, T_r$, where $v'_r = z_{crit} \, . \, v_r$ [1] and the index r indicates the "reduced" value with respect to the critical state, namely $p_r = \frac{p}{p_{cr}}$, $v_r = \frac{v}{v_{cr}}$, $T_r = \frac{T}{T_{cr}}$ and the "compressibility" z defining the deviation from "ideal gas", $pv = z \cdot RT$.

The well known curves $p_r = f\ (v_r\ ,\ T_r)$ shown in fig. I are transformed into a form with double logarithmic scale: $\log p_r = f\ (\log v_r\ ,\ \log T_r)$.

First the effects of this coordinate transformation for an ideal gas are shown strongly simplified in fig. II. From that we can easily sea that:

a) The "three dimensional hilly topography" becomes a simple inclined straight slope, wherein "topographic" anomalies can be seen immediately.

b) The isothermals, formerly hyperbelas, become straight parallel lines, with even distance inbetween them if their numeric scale is also logarithmic. Their directions become 45° when both scales are selected the same.

c) Formerly narrow bundled lines are comfortably stretched out and the same relative accuracy holds for the whole range, although this range may be enlarged to higher pressures and bigger volumes.

d) Equal ratios of pressure, volume and temperature are represented by equal distances.

e) A polytropic path is determined by the polytropic exponent n defining its straight "compass-direction". The inclination along this path leads to changes in temperature.

f) Unfortunately areas in this diagram do not represent the work anymore.

C) Now we will discuss the characteristics of the real gas in these logarithmic coordinates. The changing directions of the isothermals in fig. III, modulating the former simple slope, reveal three main ranges:

a) At large volumes (v_r larger than 5 to 10) deviation from ideal gases are very small for all temperatures (excluding the very neighbourhood of the saturation line). The molecules are hardly influencing each other: $z \simeq 1.0 = \text{constant}$.

b) Between pressures of $p_r \sim 0,2$ to 5 and temperatures T_r below 5 there exists a "flat balcony" in the slope. Volumes v_r in this region are smaller than in the "ideal" state due to attracting molecular forces: $z < 1.0$.

c) At high pressures, p_r greater than ~ 5 the "slope" at all heights, i.e. at all temperatures, is "turning". This is the region where the volumes v_r tend to keep bigger, due to the proper volume of the molecules, or their repulsive forces: $z > 1.0$.

D) Some typical topics: (note the different samples drawn into the fig. III!)

a) According to the definition the ratio of T_{ideal}/T_{real} at any point represents z. Also the ratio of the volumes v_{re}/v_{id} at the same pressure and temperature, or the ratio of pressures p_{re}/p_{id} at the same volume and temperature represents z. The value of these ratios z is again a well defined interval - the same throughout the diagram. It can easily be measured at the logarithmic scale.

b) At the crossing between a "real" and an "ideal" - Isothermal, z equals unity. But isothermal behaviour is entirely different from "ideal": The value of z on its own is not representative.

c) The direction n_T, defining any short path along a real isothermal (pv^{n_T} = const) is given by:

$$n_T = -\left(\frac{\partial \log p}{\partial \log v}\right)_T = -\frac{v}{p}\left(\frac{\partial p}{\partial v}\right)_T$$

This may be called the isothermal elasticity modul: "$n_T = \gamma$ (or $n_T = \frac{1}{y}$ in the nomenclature of Schultz. [2])

The values of n_T can be found by comparison of direction in the figure. However they could be plotted directly, as has been done for $n_T = 1$. On this particular curve all isothermals have the same direction as the ideal ones, 45° in our graph.

d) The direction for all "real polytropics" must be "turned" in a similar way as for the real isotherms.

In the special case of an isentropic path just the same turning factor n_T must be applied.

$$n_{s_{id.}} = (c_p/c_v) = \varkappa_{id}$$

$$n_{s_{real}} = n_T \cdot (c_p/c_v) = n_T \cdot \varkappa_{re} = K$$

Three important remarks must be noted here:

1) Except for monomolecular gases with c_v = constant, the value of $\varkappa_{id} = {}^c p/c_v$ also varies for "ideal gases", because c_v (and c_p) rises with rising temperature according to higher terms of molecular vibration. However $\varkappa_{id}$ is independent of pressure and the difference of $c_p - c_v = R$ holds true for all conditions.

2) In the expression of $K = n_T \cdot \varkappa_{re}$
$\varkappa_{re}$ is the true value for the real condition.
It differs from $\varkappa_{id}$.
Here the knowledge of $\varkappa_{id}$ throughout the temperaturerange, and for $\varkappa_{re}$ also throughout the pressure range, is presumed. These values cannot be caught into the system of corresponding states!

3) Designation of ${}^c p/c_v$ unfortunately differs in literature:

${}^c p/c_v$	: $\varkappa$!	n_s: K !	in modern German lit.
c_p/c_v	: K	n_s: –	other authors, Schultz.

This leads to considerable misunderstanding.

For any real polytropic, other than the isentropic, but assuming to be adiabatic and to be defined by its efficiency $\eta = \frac{v\,dp}{dh}$ (for compression) an additional correction α must be applied, which is identical with the slope along the direction of p = constant:

$$\alpha = \left(\frac{\partial \log v}{\partial \log T}\right)_p = \frac{T}{v}\left(\frac{\partial v}{\partial T}\right)_p \qquad (= x + 1 \quad \text{of Schultz})$$

Whereas the polytropic exponent for ideal gas is given by:

$$n_{id_\eta} = \varkappa_{id} \cdot \frac{1}{\frac{1}{\eta} - \varkappa_{id}\left(\frac{1}{\eta} - 1\right)} \qquad \left(\text{by transformation of } \frac{n-1}{n} = \frac{\varkappa - 1}{\varkappa \cdot \eta}\right)$$

the polytropic exponent for real gases, applying the two corrections becomes:

$$n_{re_\eta} = n_T \cdot \varkappa_{re} \; \frac{1 \quad + (\alpha - 1)}{\frac{1}{\eta} - \varkappa_{re}\left(\frac{1}{\eta} - 1\right) + (\alpha - 1)}$$

derivation see [2]

Some values of α are plotted in the diagram, so that with this one sheet only, all information on real polytropic behaviour (except $\varkappa$) can be obtained.

Alternatively the correction α could be substituted by one defining the slope along v = const: β (= L of Schultz)

$$\beta = \left(\frac{\partial \log p}{\partial \log T}\right)_v = \frac{T}{p}\left(\frac{\partial p}{\partial T}\right)_v$$

f) The temperature rise for the ideal gas is calculated by $\frac{T}{T_1} = \left(\frac{p}{p_1}\right)^{\frac{n-1}{n}}$.
It is easily seen that this does not hold for a real gas: Along a real isothermal with $n_T \neq 1$ the above expression would result in a temperature change, which is in contrast to the definition of an isothermal.
Therefore the obvious correction $\frac{Z_1}{Z_2}$ must be applied:

$$\tau_{re} = \frac{T}{T_1} = \frac{Z_1}{Z}\left(\frac{p_2}{p_1}\right)^{\frac{n_T-1}{n_T}} = \frac{Z_1}{Z}\,\pi^{\frac{n_T-1}{n_T}}$$

This holds for all polytropic temperature changes

$$\tau = \frac{Z_1}{Z}\,\pi^{\frac{n-1}{n}}$$

g) Up to now we have allways considered paths with a constant direction n. However often a compression occurs along a distance within which the exponent n changes either by change of the value n_T or by a change of $\varkappa$ or the efficiency η. Here the practical problem is to select the adequate mean values of n.
This mean value of n should yield the correct endtemperature and also the correct work. This however is not possible because between the straight direction line n leading to the correct temperature and the actual curved polytropic path lies always an area which represents an error in the amount of work.

To obtain theoretically correct means needs path integration [3]. However simple arithmetical means $\bar{n} = \frac{n_1 + n_2}{2}$ will deliver good approximation for most practical application. Looking at the curvature of the isothermals in our diagram the importance or possible neglection of such means can easily be estimated!

If the value of $\varkappa_{re}$ is not readily available a very rough approximation to calculate work consists of replacing z_1 by the mean value $\bar{z} \sim \frac{z_1 + z_2}{2}$:

$$h_2 - h_1 = \bar{z} \, R T_1 \frac{\varkappa_{id} \cdot \eta}{\varkappa_{id} - 1} \cdot \left(\pi^{\frac{\varkappa_{id} - 1}{\varkappa_{id} \cdot \eta}} - 1 \right)$$

(This is not applicable for the region around $n_{T_{re}} = 1$ as can be seen easily from the shape of isothermals in this region)

E) So far we have only spoken of "thermal behaviour", that is of functions of p,v and T. And this in generalised terms for all gases. It is only by the introduction of $\varkappa = {}^{c}p/c_{v}$ that calorific values have entered our problems. But it is through this value that the difference in individual gases have actually re-entered our system of corresponding states.

F) In our last fig.IV two typical samples of application are sketched in:

a) High-pressure compression of synthesis-gas for ammonia-production with its end pressure of 300 atm leading to the region of $z > 1{,}20$ and $n_T > 1{,}25$. This together with a rise of $\varkappa$ from 1,4 to 1,47 results in an isentropic direction of $n_s \simeq 1{,}25 \cdot 1{,}47 = 1{,}83$ (instead of $n_s = 1{,}4$ at low pressure). This of course can by no means be neglected any more.

The steady increase in the polytropic-slope from $n_1 \sim 1{,}7$ in the first casing to $n \sim 2{,}8$ in the last casing is however partly caused by the decreasing efficiency in the very small wheels at the high pressure end.

b) Compressors in a cooling cycle for liquefaction of natural gas using a mixture of different hydrocarbon gases as a refrigerant.

This process lies in the other region of real gas characteristic with $z < 1,0$. Here especially the neighborhood of the saturation line and the pressure approaching the critical one is effecting the result: $z_2 \sim 0,88$ $n_{T_2} \sim 0,87$ $n_s \sim 1,08$ (instead $n_s \simeq 1,2$ at low pressure).

With the vast volumes to be handled asking for axial blowers, it is in this case especially important to know exactly the correct temperature and with it the correct volume characteristic along the blading stages.

To conclude we may state:

The diagram discussed above allows a quick survey and an easier understanding of the effects caused by real gas behaviour. In a single sheet all influences on polytropic behaviour - with the exception of $\varkappa = {}^{c}p/c_v$ - over a wide range can be taken out with an accuracy that is restricted by the general accuracy of the laws of corresponding states. For higher accuracy detailed analysis has to be performed.

[1] Beattie, Stockmayer: Repts.Progr. in Phys. 7, 195 (1940)

[2] Schultz: The polytropic analysis of centrifugal compressors ASME 60-WA-296, 1961.

[3] Dibelius: Kann man Zustandsanderungen sicherer berechnen? BWK 15, Feb. 1963.

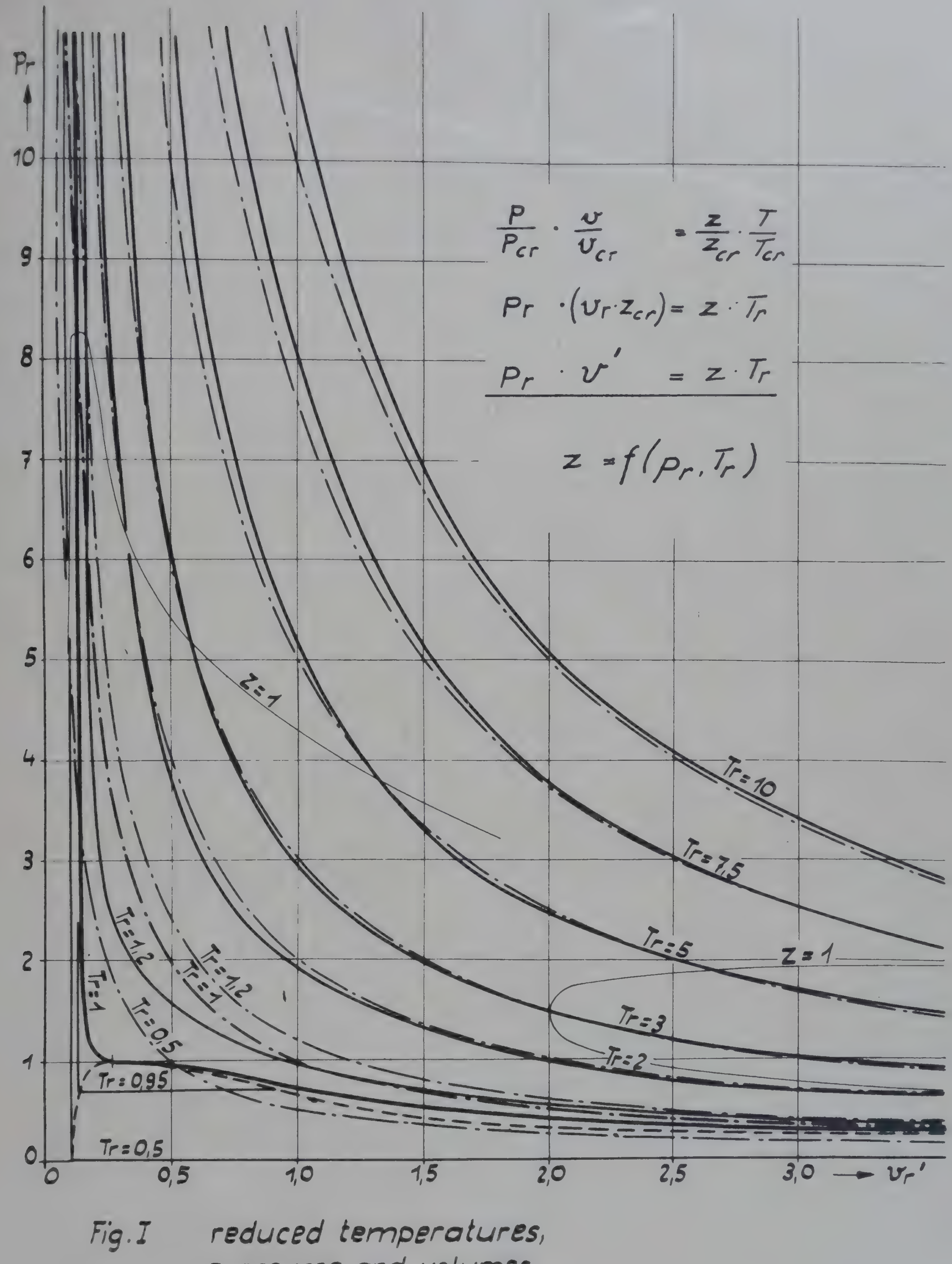


Fig. I reduced temperatures,
pressures and volumes
for —·— ideal Gases
—— real Gases

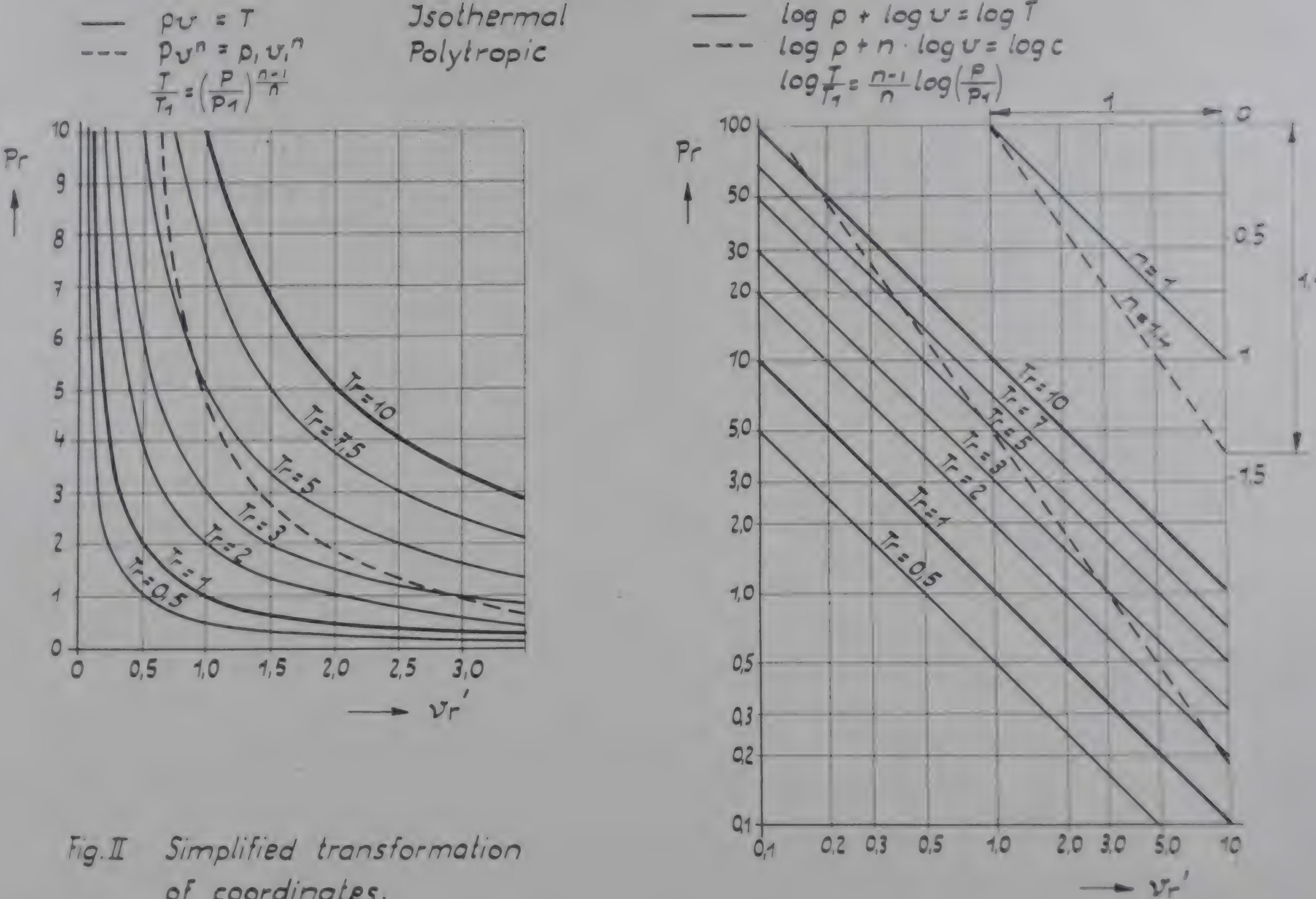


Fig. II Simplified transformation of coordinates.

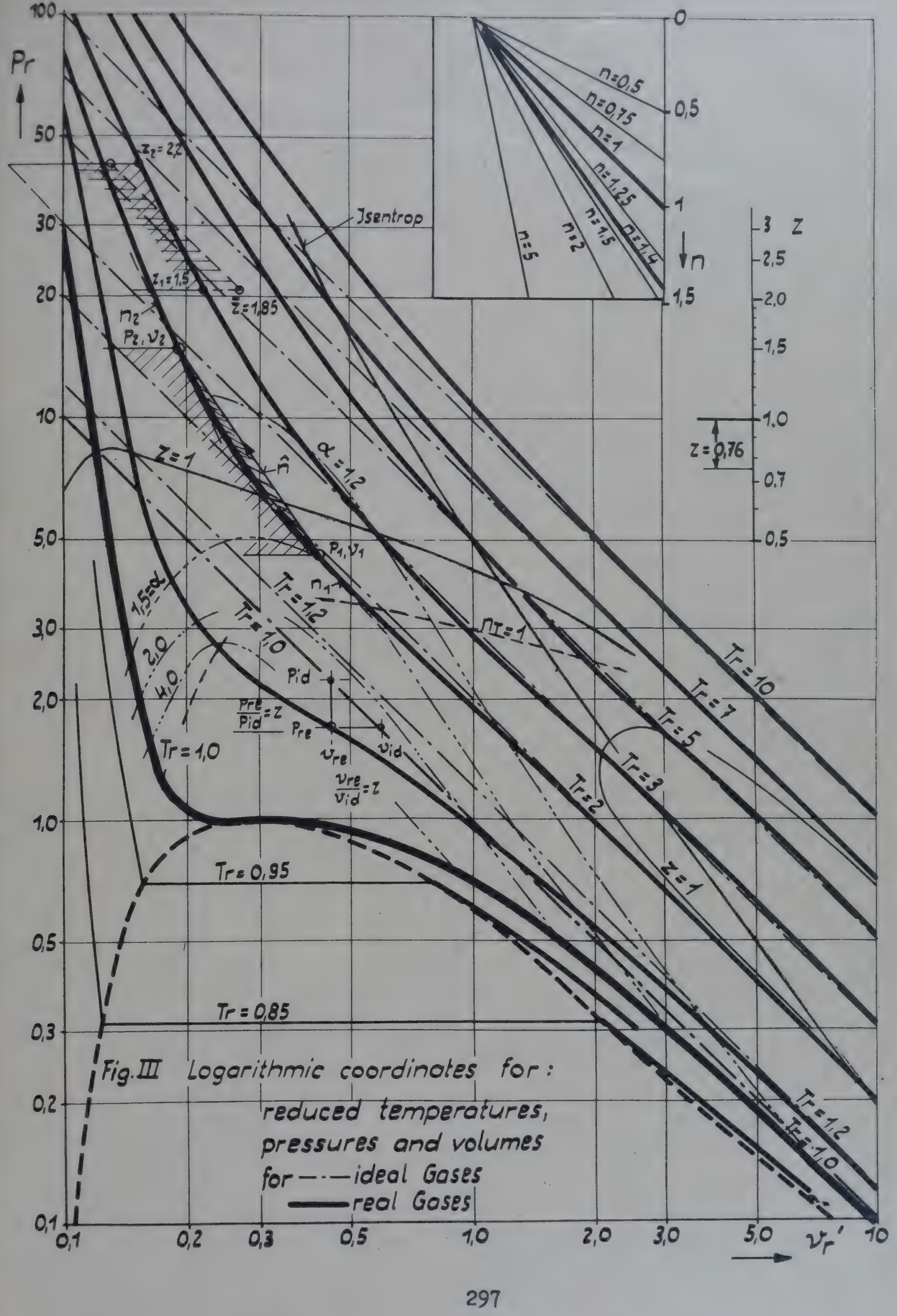


Fig. III Logarithmic coordinates for: reduced temperatures, pressures and volumes for —·—ideal Gases ——real Gases

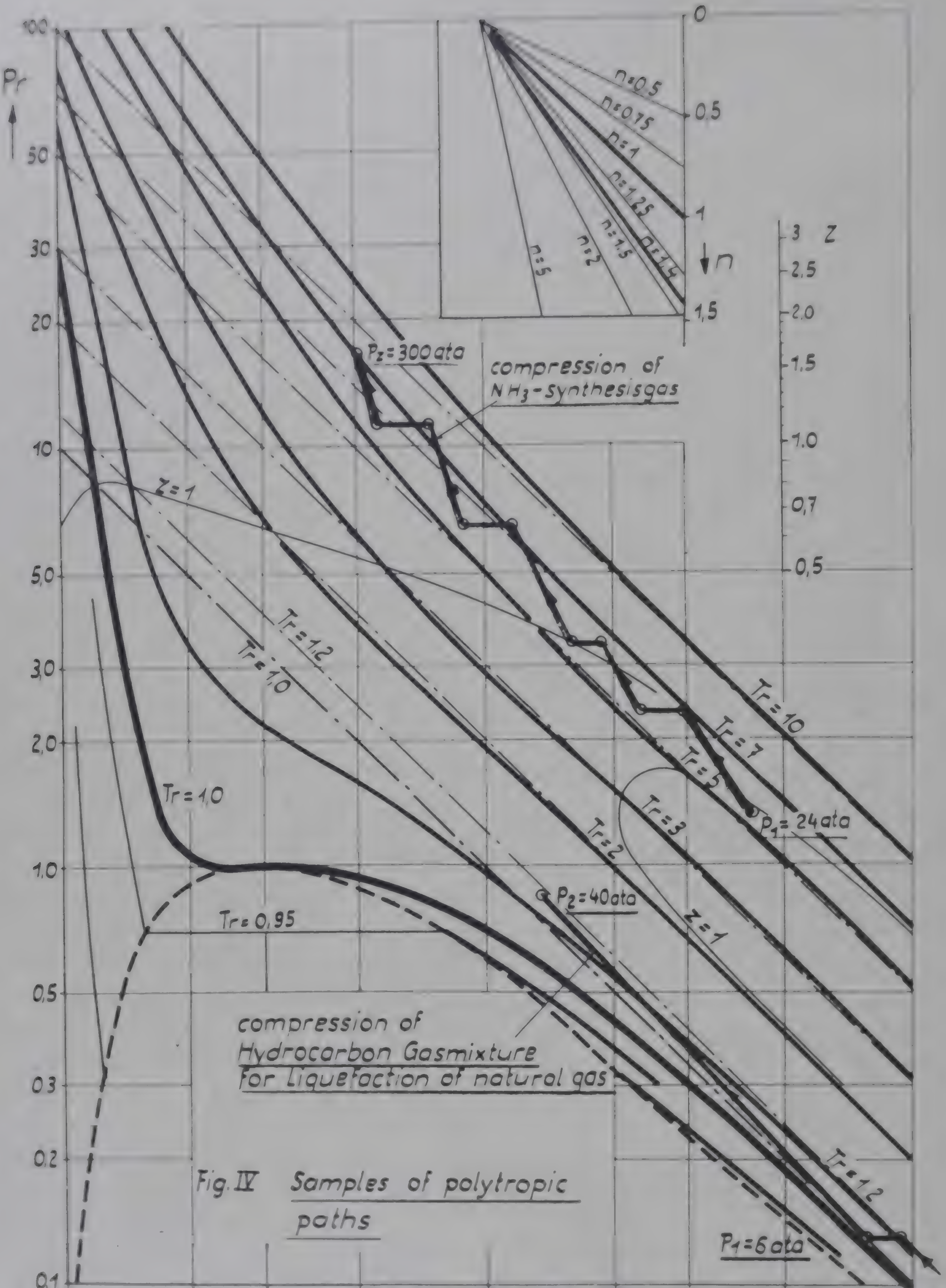


Fig. IV Samples of polytropic paths

Table

of different paths of thermodynamic states
expressed by the polytropic volume exponent: n

	ideal	real
① General path: $pv^n = \text{const}$	n	n
② Isothermal path $T = \text{const}$	$n_T = 1$	$n_T = -\left(\frac{d \log p}{d \log v}\right)_T$
③ Isentropic path $S = \text{const}$	$n_s = \varkappa\left(= \frac{c_p}{c_v}\right)$	$n_s = n_T\ \varkappa$
④ Isenthalpic path $H = \text{const}$	$n_H = 1$	$n_H = n_T \cdot \varkappa \frac{1 + (\alpha - 1)}{\varkappa + (\alpha - 1)}$
⑤ Polytropic (and adiabatic) path of compression: $\eta_{compr} = \frac{v \cdot dp}{dH} = \text{const}$	$n_\eta = \varkappa \cdot \frac{1}{\frac{1}{\eta} - \varkappa\left(\frac{1}{\eta} - 1\right)}$	$n_\eta = n_T \cdot \varkappa \frac{1 + (\alpha - 1)}{\frac{1}{\eta} - \varkappa\left(\frac{1}{\eta} - 1\right) + (\alpha - 1)}$

DESIGN OF BASE-LOAD EVAPORATORS FOR LNG

By J. K. Guthrie and E. J. Gregory*

1. This paper describes the two types of LNG evaporator designed and manufactured by Marston Excelsior Limited, namely the Open Rack or Panel type, and the Intermediate Fluid or Indirect type. Panel type evaporators are in use in the United Kingdom and France, and are currently being installed in Japan. An Indirect type is in use in the United Kingdom in parallel with the Panel type.

2. Briefly, the Open Rack or Panel type of evaporator consists of aluminium panels hanging vertically from a steel structure. Water is pumped up to the top of the panels and runs down each side of the panels, giving up heat to the LNG flowing upwards inside the panels. The water is collected in a pond beneath the panels and is returned to the source, usually an estuary or open sea.

   The Indirect type of evaporator makes use of a suitable fluid as the heating medium between natural water and the LNG. Propane is used in the Marston design, but other hydrocarbon liquids can be employed. The water is passed through a tubular heat exchanger and gives up its heat to the propane, boiling it. The propane vapour condenses on the tubes of another heat exchanger, thus heating the LNG inside the tubes. Both the heat exchangers are contained within the same vessel. This plant can be used for the generation of electric power by the expansion of a portion of the propane in a turbine, with condensation against LNG, but at present there is no economic justification for this step.

3. Working pressures for these plants vary from 75 bars to 7 bars, and designed outputs from 57 tons per hour to 16 tons per hour of LNG. It will be seen that the design allows considerable variation to suit the conditions required.

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* Marston Excelsior Limited

In the Panel Type of plant the panels are connected together in convenient groups, each group having its own isolation valves, so that sections of the plant may be shut off during periods of low load or for maintenance.

4. Panel Type Plants

The main elements in these plants are the panels themselves, which form the heat transfer elements. They are made up of numbers of extruded hollow sections arranged side by side to form a panel and tack-welded together to confer some rigidity.

The extruded tubes are expanded and welded into semicircular header extrusions at top and bottom, after which the semicircular header is closed by means of another semicircular extrusions to form a circular tube. The longitudinal welds are machine-made by the metal-arc inert gas process.

Finally, the suspension brackets for hanging the panels from the steel structure, and the flanges for gas inlet and outlet are welded.

The mechanical design of the panels follows orthodox lines, stress values for the aluminium material being obtained from the A.S.M.E. Code, Section VIII, and design formulae for calculations from the A.S.A. Code for Petroleum Refinery Piping, where applicable.

Flanges on the panels, and all other flanges on gas piping are designed to the A.S.M.E. or Taylor Forge method, now incorporated in British Standard 1515. Flanged joints are of the tongue and groove type with non-metallic gaskets.

After fabrication the panels are sprayed with a sacrificial coating of pure aluminium with about 1% zinc. This coating is limited to 0.010 inch (0.25 mm) thickness, as any greater thickness causes spalling due to thermal stresses across the coating. In practice a thickness of 0.004 inch (0.1 mm) is aimed at.

Aluminium is generally suitable for panel type plants, but to satisfy users and some metallurgists the sacrificial coating has been applied to all plants using sea-water.

The design of the bolted joints for the low-temperature flanged joints follows the method developed by Usher of I.C.I. Limited, Agricultural Division, with modifications made in the light of experience.

For most of the bolted joints, bolts of 9% nickel steel are used, with flanges of aluminium or stainless steel. For those joints on the panels or near the panels where there is a danger of corrosion from sea-water splash, aluminium alloy bolts of alloy BS.1476-HE30-WP (ASTM 6351), with sleeves of aluminium alloy BS.1471 NT 5 (ASTM 5154) are used.

The principle is that the bolt is provided with a sleeve made of a material with a negligible coefficient of thermal expansion, such as a 36% nickel/iron alloy. As the joint cools down, the flanges cool more quickly than the bolts, so that the bolt load is relieved. In the case of steel bolts with aluminium flanges, the sleeve has the effect of lengthening the bolt so that it contracts as much as the flanges, thus keeping the joint tight after steady conditions have been reached.

In the case of joints with aluminium bolts, sleeves, and flanges, the sleeve and bolt contract at the same rate, so that the effective bolt length is that within the flanges, and this is allowed for in the design.

It has been found that empirical formulae for tightening torques give torques too low for effective sealing. A better method is to measure one of the bolts accurately before tightening and then check the amount of stretch after tightening. If the extension is less than the calculated amount, the bolts must be further tightened in sequence until the required extension is attained.

Bolt stresses are as high as 25,000 lb/sq. in. (17.6 Kg/sq.mm) tensile for 9% nickel steel bolts and 8,000 lb/sq. in. (5.6 Kg/sq.mm) for aluminium bolts.

The design of the low-temperature pipework calls for no special comment, except that provision must be made for the contraction due to cooling down. This is catered for by installing stainless bellows units between the panel inlet manifolds and the branches on the inlet pipe. The bellows do not shorten and lengthen like a concertina, but move sideways as the LNG inlet pipe contracts. To allow for the contraction of the inlet branches, the inlet pipe is mounted on a support which can slide inwards towards the panels but not longitudinally.

The LNG inlet pipe and branches right up to the panel flanges is insulated by some 6 inches (150 mm) thickness of low-temperature lagging. This can be expanded ebonite or expanded polyurethane of low density, except at pipe supports, where it is of higher density to take mechanical loads.

Isolation valves for the groups of panels are of the screw down type, with stainless steel bodies split at the centre-line and having ptfe seats and gaskets. These valves can be fitted with toggle gear for operation against pressure.

The panels are hung from steel I-section joints on the steel structure on blocks of laminated plastic material to act as electrical insulators for the prevention of corrosion.

The lower ends of the panels hang free, but at the end of the lower header remote from the inlet flange the panels are tied together in groups of 4; or 5 as convenient. The tie-bar is provided with a tongue which can slide in a guide allowing the panels to contract lengthways and upwards, but not to move sideways.

The top headers of the panels are also tied together to prevent lateral movement.

These fixings do not help the panels to resist wind loads, which must be resisted by separate barriers. The steel stucture is designed to take wind loads so that wind barriers may be fixed to it. These are usually corrugated sheets of asbestos or acrylic material.

Water supply to the panels is by rubber-lined mild steel piping to aluminium distributor pipes and spray pipes. The spray pipes discharge into aluminium troughs, one on each side of each panel, and from the troughs the water flows down on the surface of the panel, into the pond below.

Regulation of water supply can be effective by a control valve linked to the gas output if it is desired to keep water flow and gas flow in step. Some operators however, prefer to keep the water flowing at full rate and accept the consequent variation in gas outlet temperature as gas flow changes. The plant works happily with either system of water control so that the choice is largely related to the cost of pumping water.

With the troughs properly adjusted to give a uniform gap between their edges and the surface of the panels the water flows downwards in a uniform film. Near the bottom of the panels the water freezes giving ice a few inches thick on the bottom header and tapering to zero thickness a few feet up the panel. The amount of ice depends on gas flow, water flow, water inlet temperature, and to a small extend on weather conditions.

The weight of ice on a panel at its maximum extend in normal operation is of the order of 1,800 lb, (820 Kg) that is, about the same weight as the panel itself.

Control of Plant

The Panel Type plant is simple trouble-free, and easy to regulate. Normal plant instruments can be used for controlling gas flow, gas temperature and water flow.

Typically, a plant has a control valve on the liquid inlet, operated from a pre-set gas flow recorder - controller on the outlet pipe. Gas outlet temperature is sensed, and operates a control valve on the water supply pipe to vary the flow of water and so give the temperature desired. Alarms are provided to shut off the gas supply in the event of failure of water supply, low gas outlet temperature or low pond water temperature.

4. Intermediate Fluid or Indirect Plants

This type of plant has a secondary fluid interposed between the LNG and the heating medium.

The Indirect Type of plant designed and manufactured by Marston Excelsior consists of a steel vessel half-filled with liquid propane at a pressure of about 4 atmospheres absolute. In the lower half of the vessel is the sea-water exchanger, and in the upper half is the LNG exchanger.

In the plant under discussion the containment vessel is of fully-killed mild steel, but having a heat-break of stainless steel to house the natural gas inlet and outlet pipes, protecting the mild steel from low temperatures from the gas piping.

The LNG exchanger itself consists of double hairpins of ½ inch bore (12 mm) Integron finned tubing in aluminium alloy BS.1476-N3.
The ends of the hairpin are expanded and seal-welded into headers of the same alloy. The whole assembly is supported on a cradle of aluminium suspended from the vessel so as to allow the heat exchanger bundle to move as it contracts on cooling.

The sea-water exchanger consists of a tube bundle of straight Integron finned tubes of 1 inch (25 mm) nominal bore arranged in 2 passes. Again, these are in aluminium alloy N3. (ASA 3003). The tubes are expanded into double tube plates of steel with aluminium facings at each end of the vessel, the whole tube bundle being supported on rollers for easy removal. As a precaution, the bores of the tubes are coated with an alloy of pure aluminium with nominally 1% zinc, as a sacrificial coating. As intended, this coating is being slowly dissolved away, giving protection to the underlying metal.

At each end of the tube bundle the adjacent surfaces of the steel and aluminium tubeplates are protected by barium chromate primer paint. Tell-tale holes are provided for the detection of leaks into the interspace between the tubeplates.
The first 4½ inches (114 mm) of tube bore and the surface of the tubeplate around the tube ends are coated with an epoxy resin as a protection against electrolytic corrosion due to the difference in potential between tube plate and tubes.

As an additional precaution, zinc anodes have been installed in the water boxes.

To allow for movements due to thermal contraction and expansion, one end of the water heat exchanger is secured to the shell by a mild stell oval bellows piece, ½ inch thick. This was made in segments, stress relieved, and radiographed.

The evaporator at Canvey Island of this type consists of two containment vessels each designed to evaporate 25 tons per hour of LNG. Each vessel is 9 ft 6 ins (3 metres) diameter by 40 ft. (13 metres) long.

Since the propane boils at 29^{O}F (minus 1.5 deg. C) the natural gas is heated to only about 27^{O}F (minus 2.5 deg. C).

To heat the gas to 0 deg. C or a little higher, tube and shell trimming exchangers are provided after the propane boilers. These exchangers are designed to take the gas in the shell and the sea-water in the tubes, for easy cleaning.

The tubes in this exchanger are of aluminium alloy BS. 1471 HT. 20 WP (ASA.6061), coated internally with pure aluminium with nominally 1% zinc, and protected at the ends with an epoxy resin, which also covers the tubeplates as for the main water propane exchanger. Zinc anodes are fitted in the water boxes as additional protection.

The shell of the trimming exchanger is a seamless aluminium tube in alloy BS 1471-N5 (ASA.5154), designed to take the gas pressure of up to 1,000 p.s.i.g. This is welded to the tube-plates of the same material.

Flanges, bolting, pipework and insulation on the Indirect plant are designed in the same way as for the panel type plant.

The plant is controlled in similar fashion to the panel type plant, namely by control of flow of water from outlet gas temperature, control of gas flow, and gas cut-offs for failure of water, low water outlet temperature, or low gas outlet temperature.

The choice of materials for the plant included 9% nickel steel, stainless steel, aluminium brass, and aluminium. 9% nickel steel and stainless steel are not sufficiently resistant to corrosion to be used for any parts in contact with sea-water, and in addition are not easily extruded. Aluminium brass is expensive and is not the easiest of metals to weld.

The choice therefore fell on aluminium as being inexpensive, comparatively easy to fabricate and to extrude, and resistant to corrosion. In addition Marston Excelsior had long experience in the fabrication of aluminium with a team of skilled welders trained in the class of work.

THERMAL DESIGN

This is a summary of the steps taken during design and since an example will clarify certain parts we shall take a requirement that 20,000 Kg/hr of L.N.G. is to be heated from - 160°C to 0°C using sea water which has a minimum winter temperature of 8°C.

The design must, of course, be based upon winter conditions.

Temperature curves

The heating curve for the L.N.G. is calculated from standard data and a typical result is shown in the accompanying graph.

The sea water enters at 8°C and, from experience, we assume an outlet temperature of 4°C as a first approximation. The cooling curve of the sea water is plotted as a straight line and the flow required follows from heat exchange requirement and temperatures.

The resulting graph is divided into sections, to enable us to take account of the various physical properties of the L.N.G. during the heating process. For simplicity five sections are shown.

From the graph the L.M.T.D. for each section is computed and hence the heat transfer requirement.

$$(hA)\ \text{req.} = \frac{Q}{\text{L.M.T.D.}} \quad \ldots\ldots\ldots\ldots\ldots\ldots(1)$$

Performance

From experience we expect to use 7 standard panels for this duty. We now calculate the (hA)ov obtained in one metre length of 7 panels for each heat exchange section as follows:-

Water Side

On this side the extrusion is a tube with two lateral fins. The extrusions are placed in line with fin tips contiguous to form a vertical surface upon which a water film can exist. This surface is now standard and the coefficients used have been arrived at first by theory and experiment, later modified by practice. The fin tips are considered ineffective and the fin efficiency is determined by normal theoretical methods. The intermediate tube surface is primary area. Thus the total effective area on this side is given by

$$\text{Aeff} = A_1 + A_2\,\text{Ef} \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (2)$$

and hence we have

$$(\text{hA})\ \text{water} = \text{h water} \times \text{Aeff} \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (3)$$

This value may be used for all sections, since the physical properties of the sea water do not vary greatly.

L.N.G. side

On this side a greater choice of tube geometry is available. However, having selected a tube geometry and taking the physical properties of the L.N.G. at the mid point of each section, the appropriate areas and coefficient may be obtained. In later approximations the boiling areas will require special consideration.

Finally for each metre of each section.

$$\frac{1}{(\text{hA})\ \text{ov}} = \frac{1}{(\text{hA})\ \text{water}} + \frac{1}{(\text{hA})\ \text{LNG}} + \frac{t}{(\text{Ak})\ \text{metal}} \quad \ldots (4)$$

and for each section

$$\text{length required} = \frac{(\text{hA})\ \text{req.}}{(\text{hA})\ \text{ov}} \quad \ldots\ldots\ldots\ldots (5)$$

A summation of these lengths gives a first approximation of the required length but this will be considerably increased by the effect of ice formation. For this effect we require the metal temperatures.

Metal temperatures

In any heat exchange system

$$(hA) = \frac{Q}{\Delta T} \quad \dots\dots\dots\dots\dots\dots\dots\dots\dots\dots (6)$$

Thus
$$(hA)\ \text{water} = \frac{Q}{(\Delta T)\ \text{water to metal}}$$

and
$$(hA)\ \text{ov} = \frac{Q}{(\Delta T)\ \text{water to LNG}}$$

Hence

$$Q = (hA)\ \text{water} \times (\Delta T)\ \text{water to metal} = (hA)\ \text{ov} \times (\Delta T)\ \text{water to LNG} \quad \dots\dots\dots\dots\dots\dots\dots\dots\dots\dots (7)$$

From this the temperature difference from water to metal and hence the temperature of the metal nearest the water follows. This temperature is then plotted on the graph. Sea water freezes at -2°C and therefore below this metal temperature ice will form. The line of ice formation is shown on the graph. To the right of this line the previous calculations are valid.

To the left of this line the performance must be re-calculated with the additional resistance to heat transfer caused by the ice. Thus formula (4) is modified to

$$\frac{1}{(hA)\text{ov}} = \frac{1}{(hA)\ \text{water to ice}} + \frac{1}{(hA)\ \text{LNG}} + \frac{(t)}{(Ak)\ \text{metal}} + \frac{(t)}{(Ak)\ \text{ice}} \quad \dots\dots\dots\dots (8)$$

The final result of the summation of the lengths required for each section yields the overall length required of the panels.

The design is then finalized by repetition of the above method with any necessary amendments.

Nomenclature used

A	Area	M^2
A_1	Primary area	M^2
A_2	Secondary area	M^2

Aeff	Effective area M^2.	
Ef	Fin Efficiency	
h	Heat exchange coefficient	Keal/hr ^oC M^2
(hA)	Heat exchanger for any fluid	Keal/hr ^oC
(hA) ov	Overall heat exchange rating achieved	Keal/hr ^oC
(hA) req	Overall heat exchange rating required	Keal/hr ^oC
k	thermal conductivity	Koal/hr.m.
L.M.T.D.	logarithmic mean temperature difference	^oC
Q	Heat exchanged	Keal/hr
t	thickness	m
ΔT	temperature difference	^oC

Subscripts are self-explanatory

In the opinion of the authors there are good prospects for the installation of more base-load LNG evaporators in various parts of the world. Interest is being shown in Japan for further imports of LNG, and in the UK there has been talk of LNG terminals in the northern half of the country.

In continental Europe, Yugoslavia, Greece and Turkey are potential users of LNG, although the advent of Russian natural gas by pipeline may delay the use of LNG in these countries. In the New World, Chile and the Argentine are seen as future importers of LNG, provided money is available to finance the projects.

If some of these evaporation facilities can be combined with users of cold, such as liquid air plants, financially-attractive industrial complexes can be planned adjacent to the LNG terminal at the sea-coast Such integrated plants will require a steady demand for gas, free of the widely fluctuating loads of domestic users. Hence the base-load evaporator should be truly a steady-load installation, supplying continuous users of gas such as industrial works, so that the supply of cold is always available.

It is clear that these ideas give scope for bold and imaginative planning.

The authors wish to thank the Directors of Marston Excelsior Limited for permission to publish this paper, and the staff of North Thames Gas Board for their help and information during the past few years.

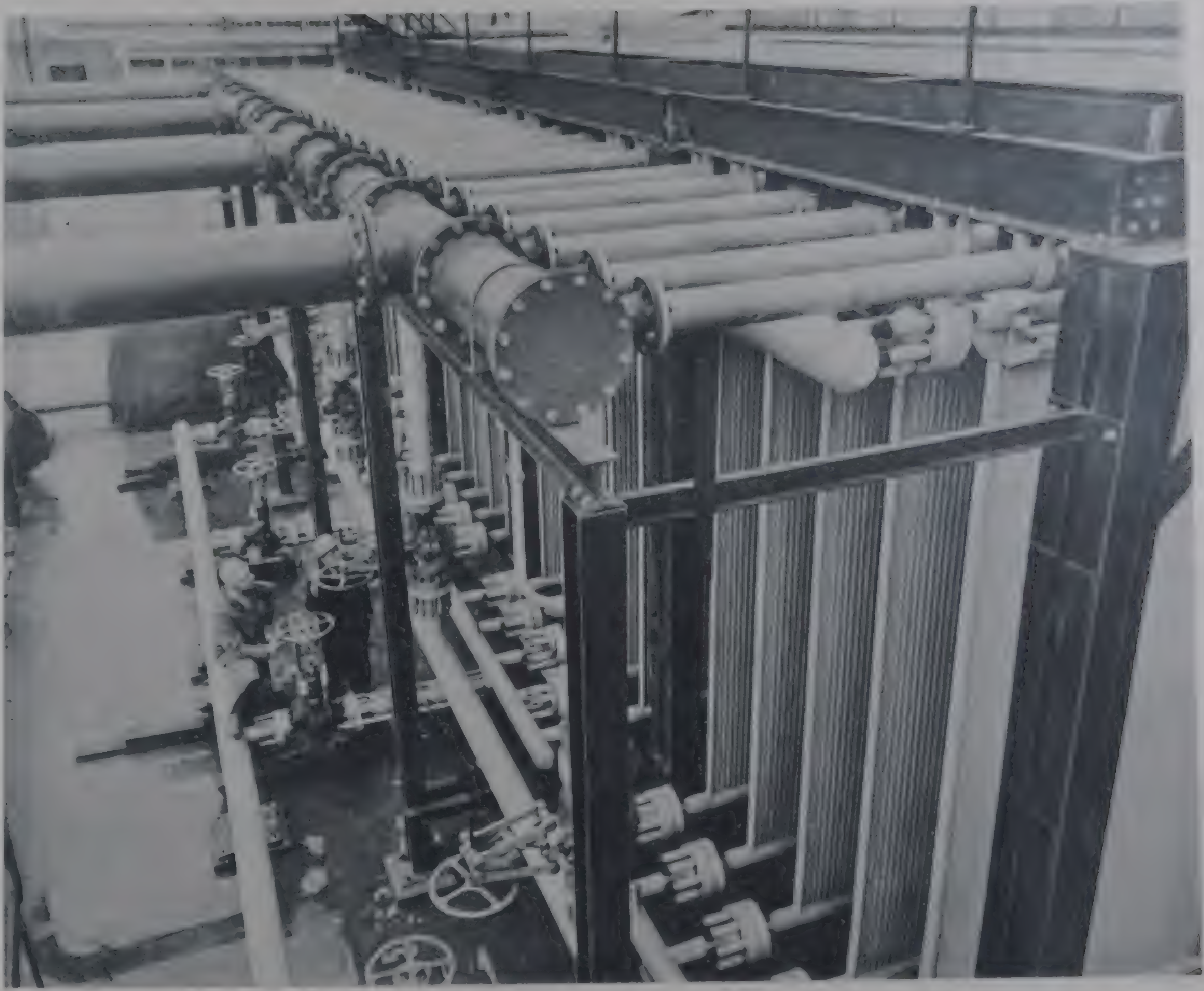


Fig.1. General view - panel type

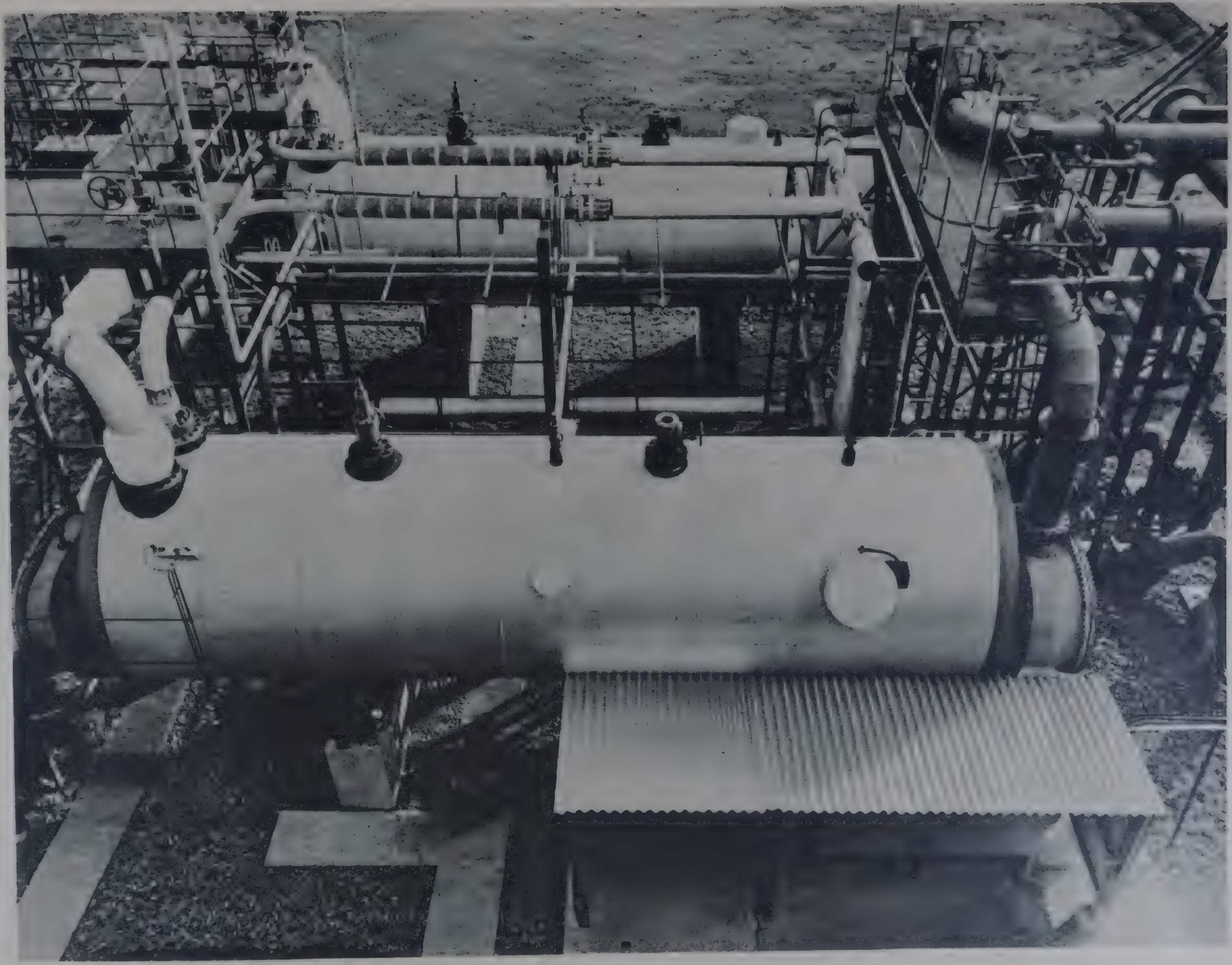


Fig.2. General view - indirect type

Fig. [illegible] panel in Marston's shops

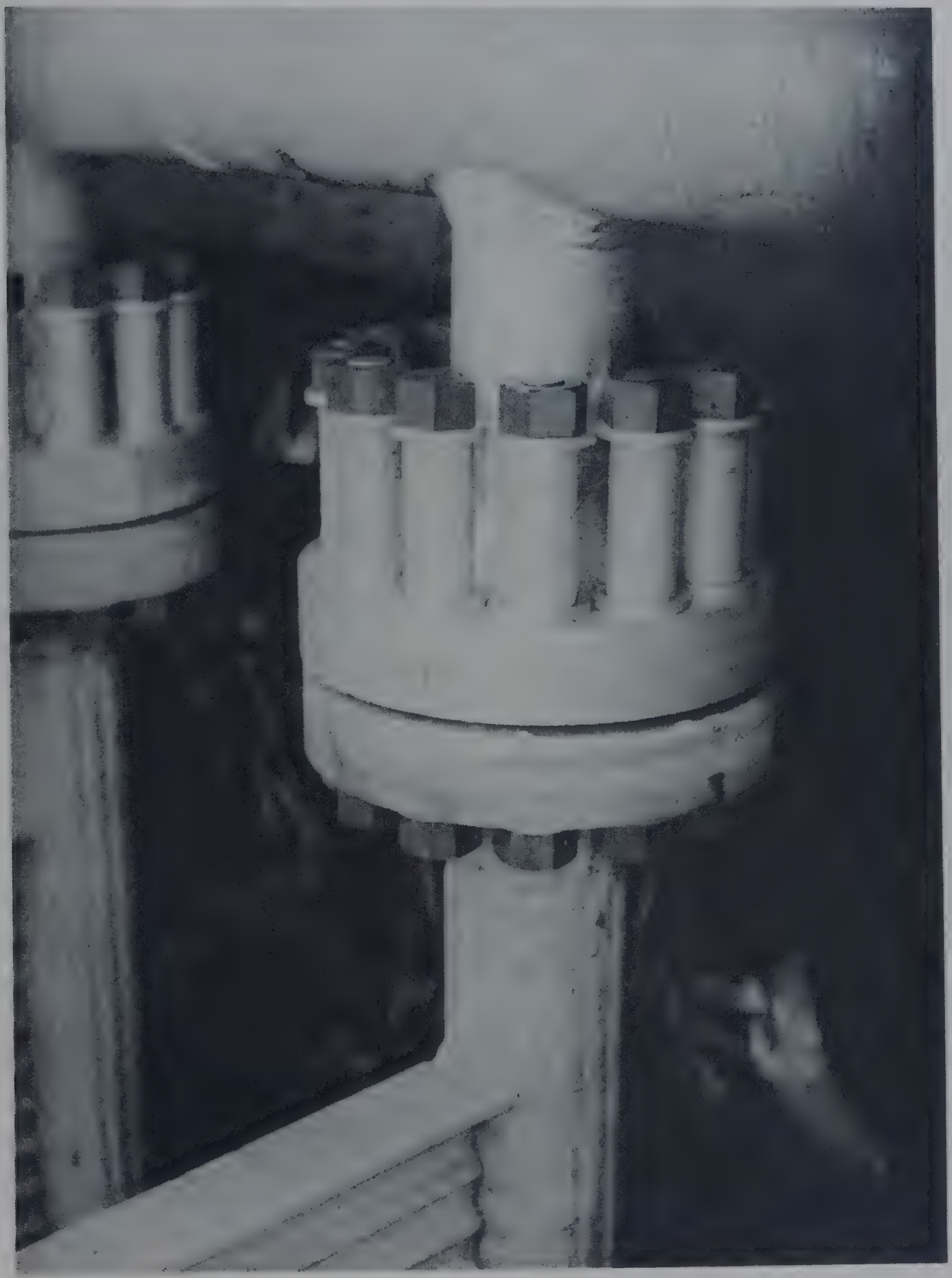


Fig.4. Low-temperature bolted joint

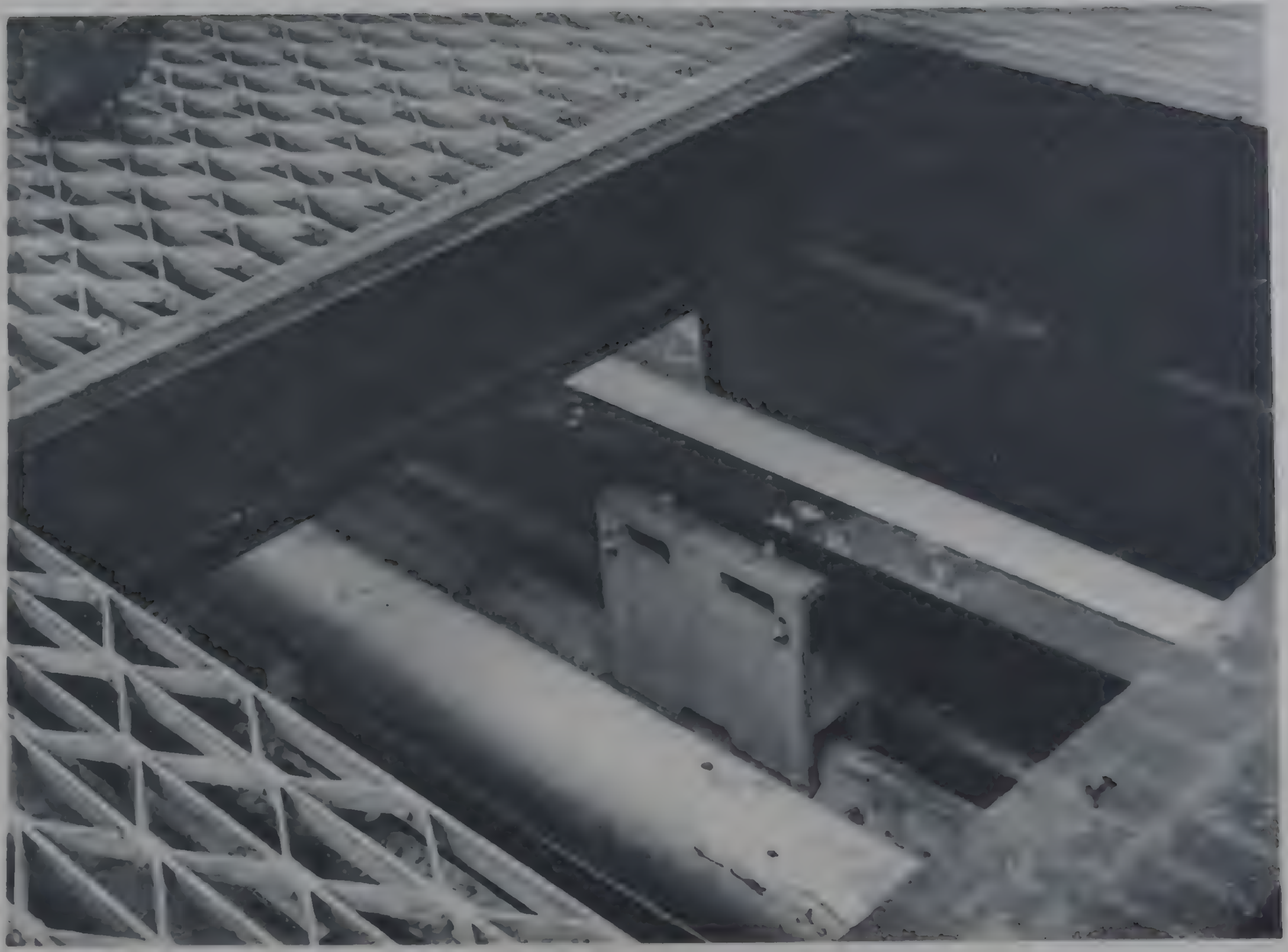


Fig.5. Suspension brackets for panel

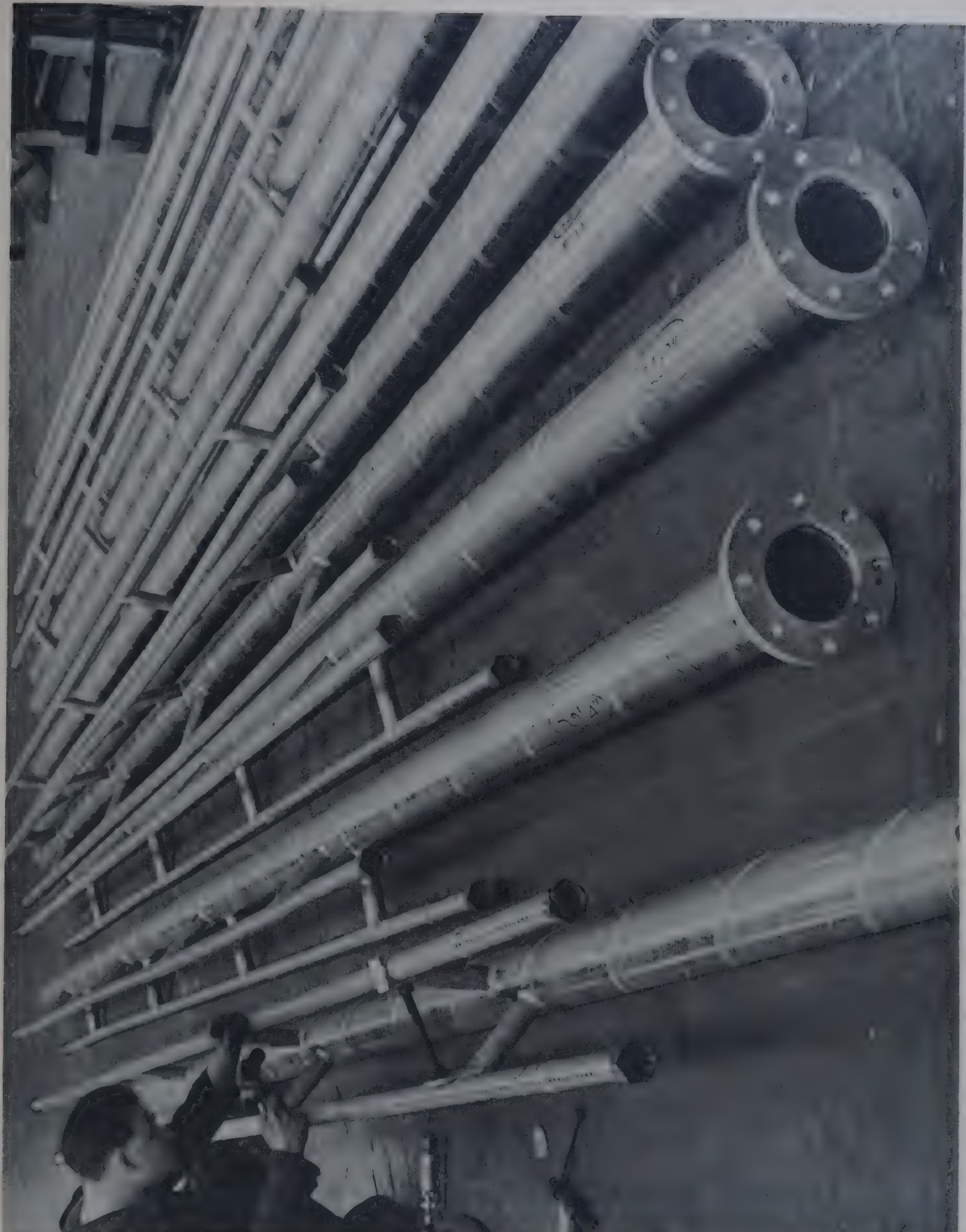


Fig.6. Water spray pipes for panels

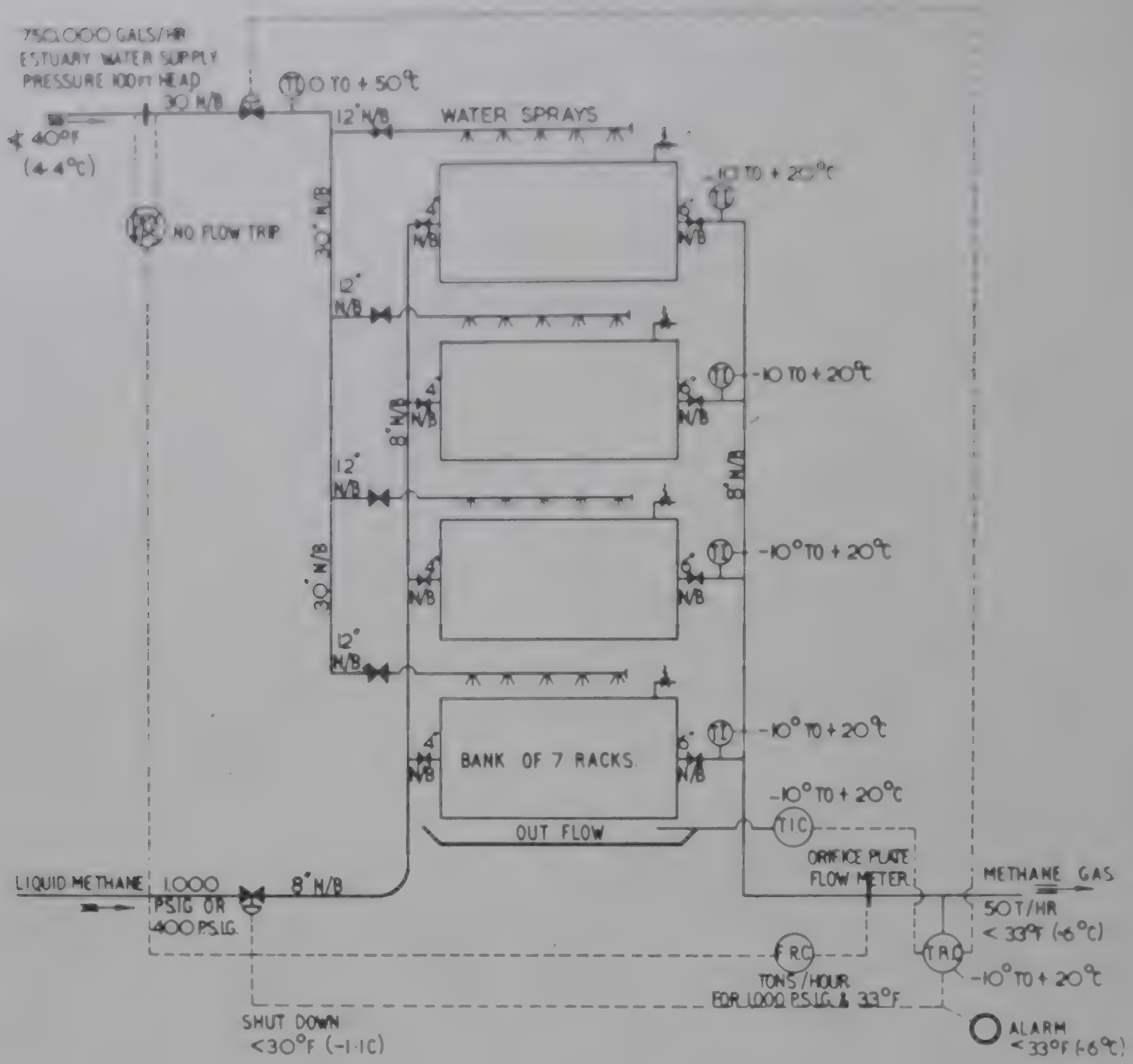
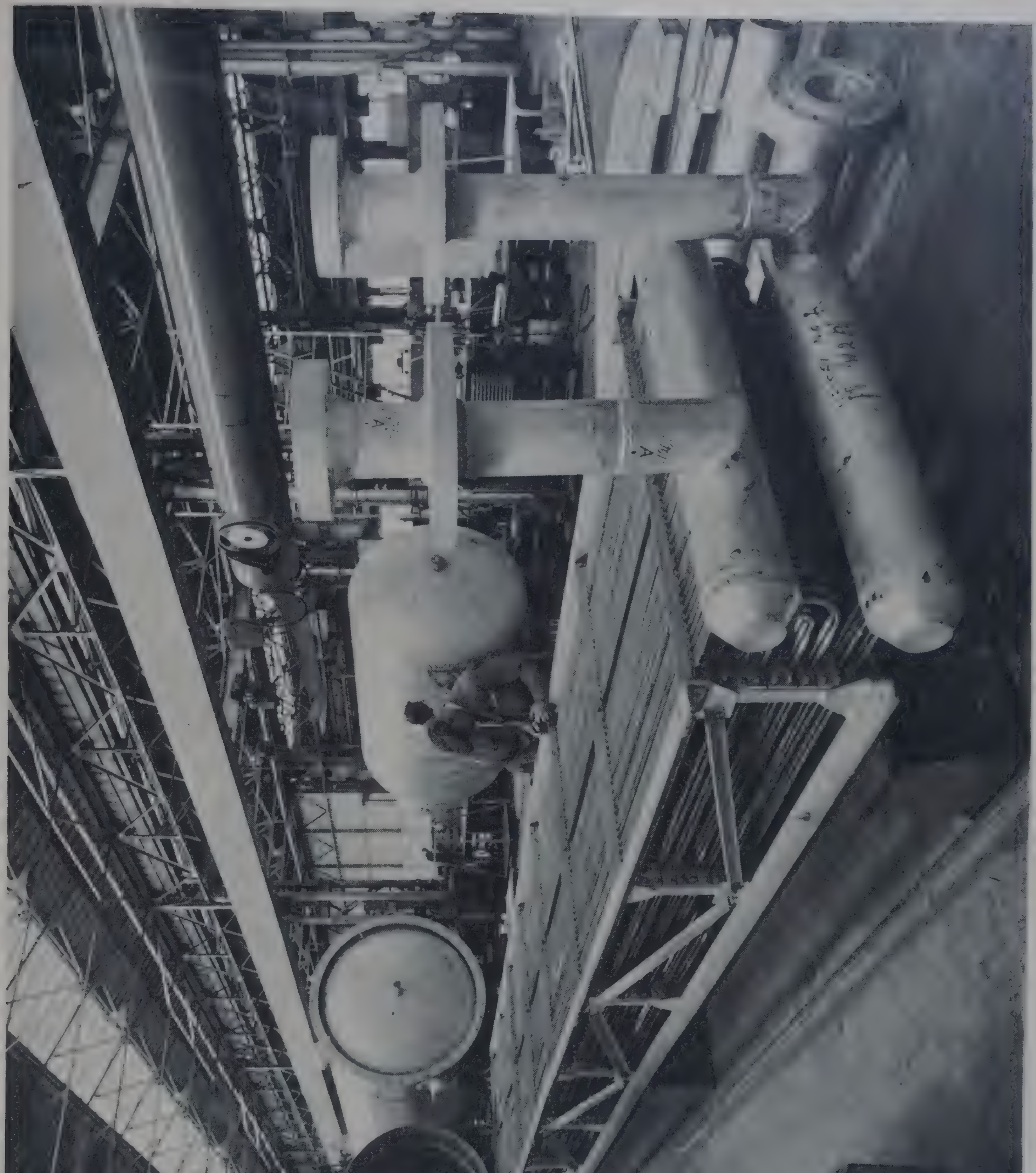

750,000 GALS/HR
ESTUARY WATER SUPPLY
PRESSURE 100FT HEAD
30 N/B
40°F
(4·4°C)
TI
0 TO + 50°C
NO FLOW TRIP
12" N/B
WATER SPRAYS
30" N/B
12" N/B
4" N/B
6" N/B
TI
-10 TO + 20°C
8" N/B
-10 TO + 20°C
-10° TO + 20°C
-10° TO + 20°C
BANK OF 7 RACKS
OUT FLOW
-10° TO + 20°C
TIC
ORIFICE PLATE
FLOW METER
LIQUID METHANE 1,000
PSIG OR
400 P.S.I.G.
8" N/B
METHANE GAS
50 T/HR
< 33°F (-6°C)
FRC
TONS/HOUR
FOR 1,000 P.S.I.G. & 33°F
TRC
-10° TO + 20°C
SHUT DOWN
<30°F (-1·1C)
ALARM
< 33°F (-6°C)

Fig.7. Control diagram - panel type



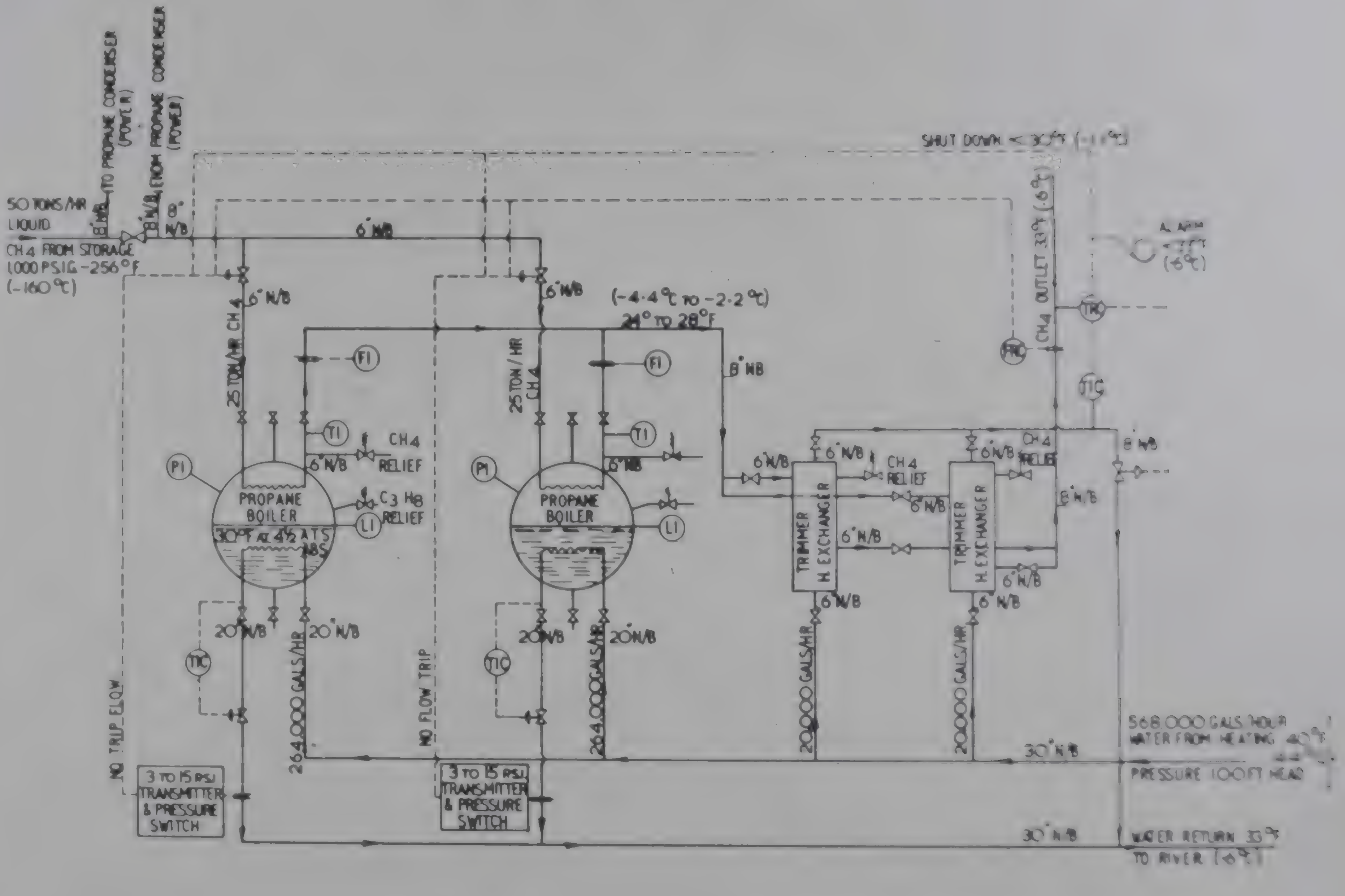

50 TONS/HR
LIQUID
CH4 FROM STORAGE
1000 PSIG -256°F
(-160°C)
8" N/B TO PROPANE CONDENSER
(POWER)
8" N/B FROM PROPANE CONDENSER
(POWER)
8"
N/B
6" N/B
SHUT DOWN < 30°F (-1°C)
6" N/B
25 TON/HR CH4
6" N/B
25 TON/HR
CH4
(-4.4°C TO -2.2°C)
24° TO 28°F
8" N/B
FI
FI
TI
TI
CH4
RELIEF
C3 H8
RELIEF
6" N/B
6" N/B
PI
PI
PROPANE
BOILER
PROPANE
BOILER
30°F AT 4½ ATS
ABS
LI
LI
20" N/B
20" N/B
20" N/B
20" N/B
TIC
TIC
264,000 GALS/HR
264,000 GALS/HR
NO TRIP FLOW
NO FLOW TRIP
3 TO 15 PSI
TRANSMITTER
& PRESSURE
SWITCH
3 TO 15 PSI
TRANSMITTER
& PRESSURE
SWITCH
6" N/B
6" N/B
CH4
RELIEF
6" N/B
6" N/B
TRIMMER
H. EXCHANGER
TRIMMER
H. EXCHANGER
6" N/B
CH4
RELIEF
6" N/B
6" N/B
6" N/B
8" N/B
8" N/B
20,000 GALS/HR
20,000 GALS/HR
CH4 OUTLET 33°F (.6°C)
FRC
TRC
TIC
ALARM
(.6°C)
30" N/B
30" N/B
568,000 GALS/HOUR
WATER FROM HEATING 40°F
PRESSURE 100FT HEAD
WATER RETURN 33°F
TO RIVER (.6°C)

Fig.9. Control diagram - indirect type

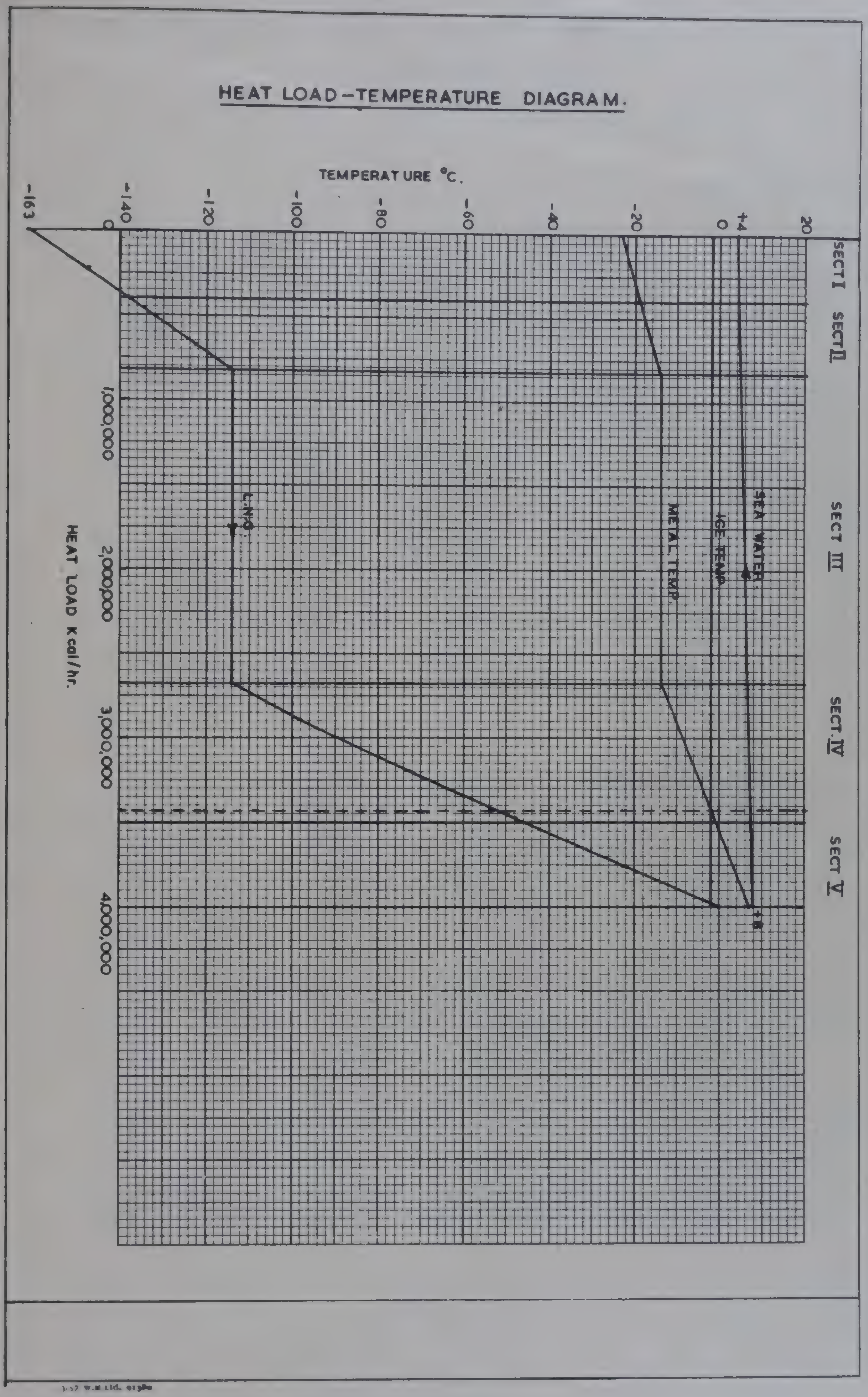


Fig.10. Heat load - temperature diagram

Development of a high performance evaporator for liquefied gases.

Dipl.-Ing. H. Waldmann

Introduction :

In evaporation processes which involve great temperature differences between the heating surface and the medium to be evaporated there forms a vapour film which impedes the heat transfer. This fact calls for very great heating surfaces which, in consideration of the steadily growing dimensions of the apparatuses, makes itself felt in many respects, such as investment costs, transport and space requirement.

In the development under consideration special attention has been paid to the improvement of the heat transfer of a high performance evaporator. A comparison with the costs of other types of evaporators will confirm the soundness of the development.

Solution of the problem of heat transfer.

a) Improvement of the heat transfer in film evaporation by means of a special type of fins.

Tests have proved how the heat transfer on the side in contact with the evaporating liquid is improved by cutting the socalled Leidenfrost vapour film which forms between the medium to be evaporated and the heated tube wall. Such cutting of the vapour film is done by swirl vanes made of wire or plate, which are connected to the tube wall in a way that assures satisfactory heat conductivity.

During evaporation of liquefied gases in the tube it could be demonstrated that with a tube of normal roughness the heat transfer on the side in contact with the evaporating liquid will be increased to 3 times the value for a normal tube, by the installation of a coil fitti to the inner tube wall.(see also [1]).
By the installation of corrugated multi-fin coils which create a flow configuration of continuous eddy current the heat transfer was improved by more than ten times. In this connection it is pointed to the fact that the Leidenfrost film may likewise be avoided if the outside or inside of the evaporator tube is so insulated that the temperature difference between the generally considerably warmer side in contact with the heating medium and the medium to be evaporated is small, so that we get bubble boiling instead of film boiling. Due to the insulation it is , however, impossible to build high performance evaporators although the heat transfer value may be double that for a normal tube [2].

b) Improvement of the heat transfer on the tube of the side in contact with the heating medium

It was only after we had succeeded in improving the heat transfer on the side of the tube in contact with the evaporating medium that it was reasonable to also make attempts to arrive at higher heat transfer coefficients on the side of the heating medium.

In the case under consideration where a technically simple solution was aimed at the objective was reached by high turbulence in the liquid bath in which the evaporator tubes are arranged.

A series of tests with a pilot plant showed increasing heat transmission values in the following sequence:

a) heating of the transfer medium with rising vapour bubbles;

b) heating with steam, and additional mechanical circulation;

c) heating with exhaust gases which agitate and circulate the bath.

c) Influence of the boiling pressure on the heat transfer during evaporation

According to Farber and Scorah (3) the h-value for film evaporation of water increases by ten times when the boiling pressure is raised with the factor 8. A similar influence of the boiling pressure on the heat transfer was found during bubble boiling of CH_4 on a smooth surface, as is shown in Table 1.

Table 1: Dependence of the heat exchange on the boiling pressure of methane.

Pressure ratio $\pi = \frac{P_B}{P_{crit.}}$	Q max $\left[\frac{kcal}{m^2 h}\right]$
0	$15,1 \cdot 10^4$
0,021	$30,2 \cdot 10^4$
0,21	$65,0 \cdot 10^4$
0,42	$68,5 \cdot 10^4$
0,63	$54,3 \cdot 10^4$
0,84	$23,6 \cdot 10^4$

From this second example of the literature studied a maximum of heat flux, Q max, is found at a boiling pressure of about 0,3 to 0,4 times that of the critical pressure (see also [3]).

It can thus be seen that a desirable reduction of the heating surface by the influences specified under a) b) and c) is possible. Although it may not always be possible to take c) into consideration on account of process requirements, the use of a special type tube for the kinds of heating described resulted nevertheless in a considerable improvement against the conventional type of tubes.

Design of the high performance evaporator

Fig. 1a, 1b, 2a, 2b

General description

The apparatus dealt with in this paper is a hot water bath evaporator with one tube bundle or more tube bundles located in a tank. The heating medium, in this present case water, can be heated either by water vapour, hot gases or air/water vapour mixtures.

Tube bundle

The special tubes made for example of copper are provided with inserts of corrugated multi-fin coils, the outside of the tubes with thin-walled coil fins arranged transverse to the tube.

A tube bundle consists for instance of 36 straight tubes in square arrangement, the maximum tube length being at present about 4 meters. Via a tube plate with dished head the liquid gas is fed to the tube system, ant the evaporated medium drawn off from the tube system. The advantage of this construction is that it is possible to tin the welded-in tube ends. Such tinning is required in the presence of acetylene in the liquid gas when using non-ferrous metals.

Equipment for the circulation of the water bath

A precondition for satisfactory heat transfer is the correct admission and great flow velocity of the heating medium which, in the present case, is water.

Steam-heated type Fig. 1a / 1b

Uniform distribution of water in the steam-heated type is obtained by feeding part of the circulated water to the tube bundle over a weir and the remainder through a sprinkler. Circulation around the deflector is due to the fact that the liquid which in a channel passes from top to bottom through the horizontal tube bundle, and which is cooled down during such process, has a higher specific weight than the liquid in the other channel which is mixed with rising vapour bubbles. (See view B, Fig. 1b). In the case of the high performance evaporator the water circulation is increased to an optimum value by sucking off part of the heated water and distributing it across the tube bundle with high energy of flow.

The steam injection, Fig. 1a / 1b, is represented as a tube equipped with nozzles.

Submerged flame-heated evaporator. See picture 2a / 2b Uniform loading of the tube bundle with exhaust gases is effected via a distribution box which is arranged parallel to the tube bundle and at a safety distance from the bottom-most range of tubes. As is the case in the steam-heated evaporator-though in inverse direction of flow- the heating medium circulates around one deflector, two deflectors respectively. The number of deflectors depends on the mnumber of bundles of which the apparatus consists.

Development of standardized high performance evaporators.

As has already been said, an evaporator consists of 1 to 4 tube bundles. Using, in addition, tubes with a length ranging from 2 - 5 meters we obtain a complete series of evaporators with uniform construction elements for capacities of a single apparatus from 1 to 20 Gcal/h.

The maximum pressure in the evaporator is 70 atü (~ 1000 psi)

The evaporators under consideration are specifically destined for peak shaving gas supply plants. Is is a characteristic of such units that they will be used for a short period of time only, but that they will have to be absolutely reliable whenever in operation.

For this reason a number of burners has been provided for the heating process, e.g. 2, 3, or 4 in order to dispose always of a heating capacity of 100 per cent should a burner happen to be out of operation. This means, however, an overdimensioning of 100, 50 or 33 per cent.

Table: 2

The series of evaporators will be standardized as follows:

Type	Heating capacity	Liquid gas flow Evaporation and heating to 20°C		
	Gcal/h	Nm^3/h in 10 000		
		O_2	CH_4	C_2H_4
I	1,6	1,14	1,03	0,8
II	2,4	1,71	1,55	1,2
III	3,6	2,66	2,32	1,8
IV	5,6	4,00	3,60	2,8
V	8,0	5,70	5,15	4,0
VI	12,0	8,60	7,75	6,0
VII	20,0	14,30	13,10	10,2

Here, it must be stressed that this design applies only to high performance evaporators with liquid gas pumps where the pressure drop plays no important role.

Operation of the evaporator and safety problems

Operation

The method of operation is described on the basis of a submerged flame-heated evaporator (see pictures 3 and 4). It is possible to evaporate certain quantities of liquid also during the heating period.

Safety problems

The plant will be shut down if the following alarms are released:

1) Water level minimum LA
2) Flame breakdown of the burners (periodical automatic supervision of the alarm system)
3) Supervision of the exhaust gas (tube burst) ARA
4) Air quantity minimum
5) Pressure in the feed line for heating medium and air.

Control

1) During continuous operation the capacity of the burners, i.e. the burning medium, is controlled by the outlet evaporated gas temperature (TIC, picture 3).

2) The air feed is so controlled that the ratio air/burning medium remains constant (FRC, picture 3).

Economy

The new developped high performance evaporator is characterized by the saving of heating surface of about 80 to 90 per cent, and a corresponding small construction unit. Compared to the normal tube evaporator this means that, even if expensive special tubes were used, the apparatus will on the whole be cheaper, the more so as the evaporator tank will also be smaller and lighter. The reduced space requirement makes itself especially felt in chemical plants where several apparatuses of identical dimensions are often installed to ensure safe production. Due to its compactness the transport and erection of a complete unit offers no difficulty.

Comparing the operating costs for the heating of different evaporators there results a distinct difference between steam-heated and submerged flame-heated types.

Relating the effeciency of the heating system to the net calorific power -to do so is normal in heating techniques- results in an effeciency greater than 100 per cent. This because of the fact that the exhaust gases are injected into the water bath and the oxidation water is not evaporated but remains in the water bath.

Operating the evaporator at a bath temperature of 50° to 60°C, heating gas can be saved due to the difference between the gross calorific value and the net calorific power:

Fuel gas	Heating gas saved
Coke oven gas	14 per cent
Natural gas	11 per cent
C_2H_6	8,4 per cent
C_3H_8	8 per cent
C_4H_{10}	7,4 per cent

Taking further into consideration that the exhaust gases leave a direct-fired equipment at a temperature of about 100 to 200°C and that there are also considerable losses due to radiation of the furnace walls, we save about 20 per cent of burning medium in case of heating by submerged flame.

For this reason it appears useful to investigate the question as to after what time natural gas burners of the submerged type will be amortized by the saving of 20 per cent of the quantity of burning gas. Such savings must, however, be set against the surplus expenses for higher compression of the combustion air.

Example: H_o = 8700 kcal/Nm3; air factor n = 1.1

Type of burner		1	2	3	4	5	6
Capacity of burner	Gcal/h	1,6	2,4	3,6	5,6	8,0	12,0
Price of burner	DM	12000	14000	16000	19000	24000	31000
Surplus cost for air compression (0.1 DM per kWh)	$\frac{DM}{h}$	0,77	1,15	1,73	2,68	3,84	5,75
Reduced expenditure due to saving of burning medium (0.1 DM per Nm3/h)		3,84	5,76	8,6	13,4	19,2	28,8
Amortization	h	3130	2440	1860	1420	1250	1080

Comparison of the costs for different types of evaporators

The worth of the new development can best be appreciated if we compare the production cost of different types of apparatus for the evaporation of a given quantity of liquid.

The cost of production is the sum of the unit costs for the evaporator, evaporator heating, control system and alarm system.

The comparison, as to the investment cost, covered the following types of apparatus:

A direct-heated tubular furnace with tubes of stainless steel

B hot water bath evaporator with normal copper tubes oil cycle evaporator
primary cycle, tube bundle of stainless steel, for oil
secondary cycle, tube bundle of copper, for liquid gas

D/E hot gas heated unit

F submerged flame-heated evaporator with tube bundle of special copper tubes.

As to the overall costs the situation is as follows: The most expensive solution is the direct heated tubular furnace (A) due to the necessity of using great quantities of stainless steel tubes.

The cheapest solution is the submerged flame-heated evaporator (F) with tubes of the corrugates multi-fin type, due to its very small heating surface and corresponding small dimensions. This price advantage applies in principle also to steam-heated evaporators with tubes of the corrugated multi-fin type. A comparison of the customary steam-heated normal tube evaporator (B) with the submerged flame-heated evaporator (F) demonstrates clearly the influence of the special type tubes and the improvement of the heat transfer on the conception of the evaporator and thus also on the price.

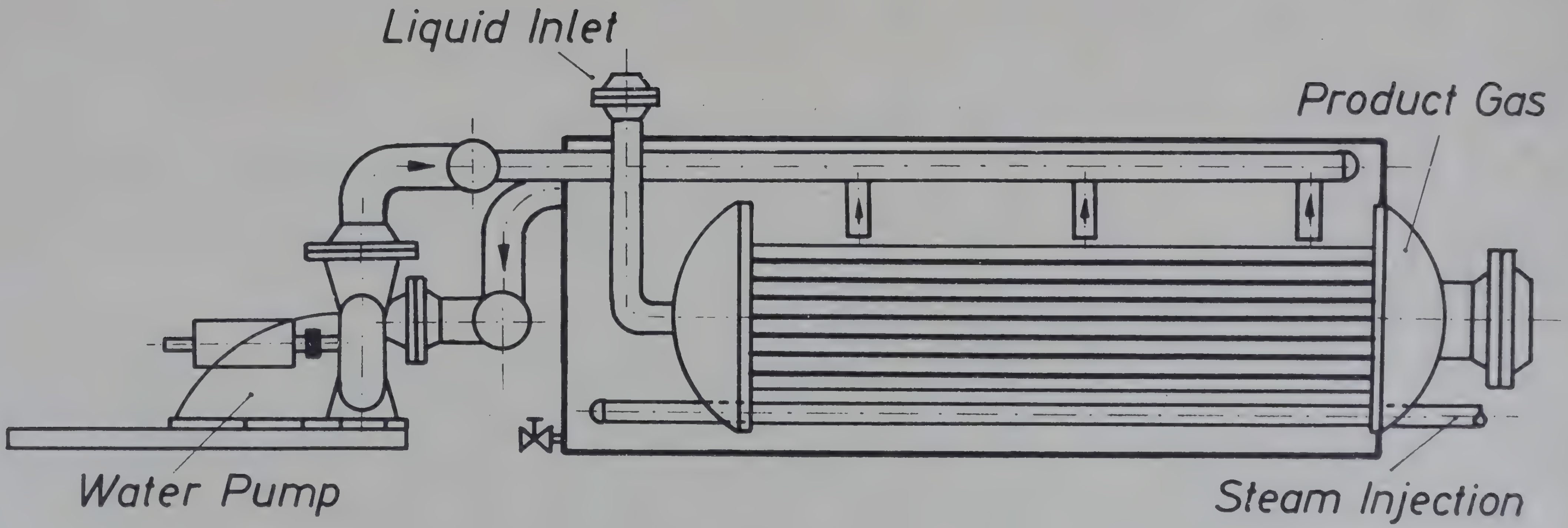
Expressed in figures, a customary type evaporator is by about 150 per cent, and a direct heated tubular furnace by about 200 per cent dearer than the new developped high performance evaporator with special tubes operating on the submerged flame technique.

Literature:

[1] Sheriff — Int. J.Heat and Mass Transfer Vol.9 p.p. 1297 - 1320

[2] Bewilogua, Knöner — Journal of the American Chemical Society / 90 : 12/ June 5, 1968

[3] Grigull — Grundgesetze der Wärmeübertragung, Springer Verlag S. 334 - 336


Liquid Inlet
Product Gas
Water Pump
Steam Injection

VIEW A

Fig.1a. Steam heated high performance evaporator

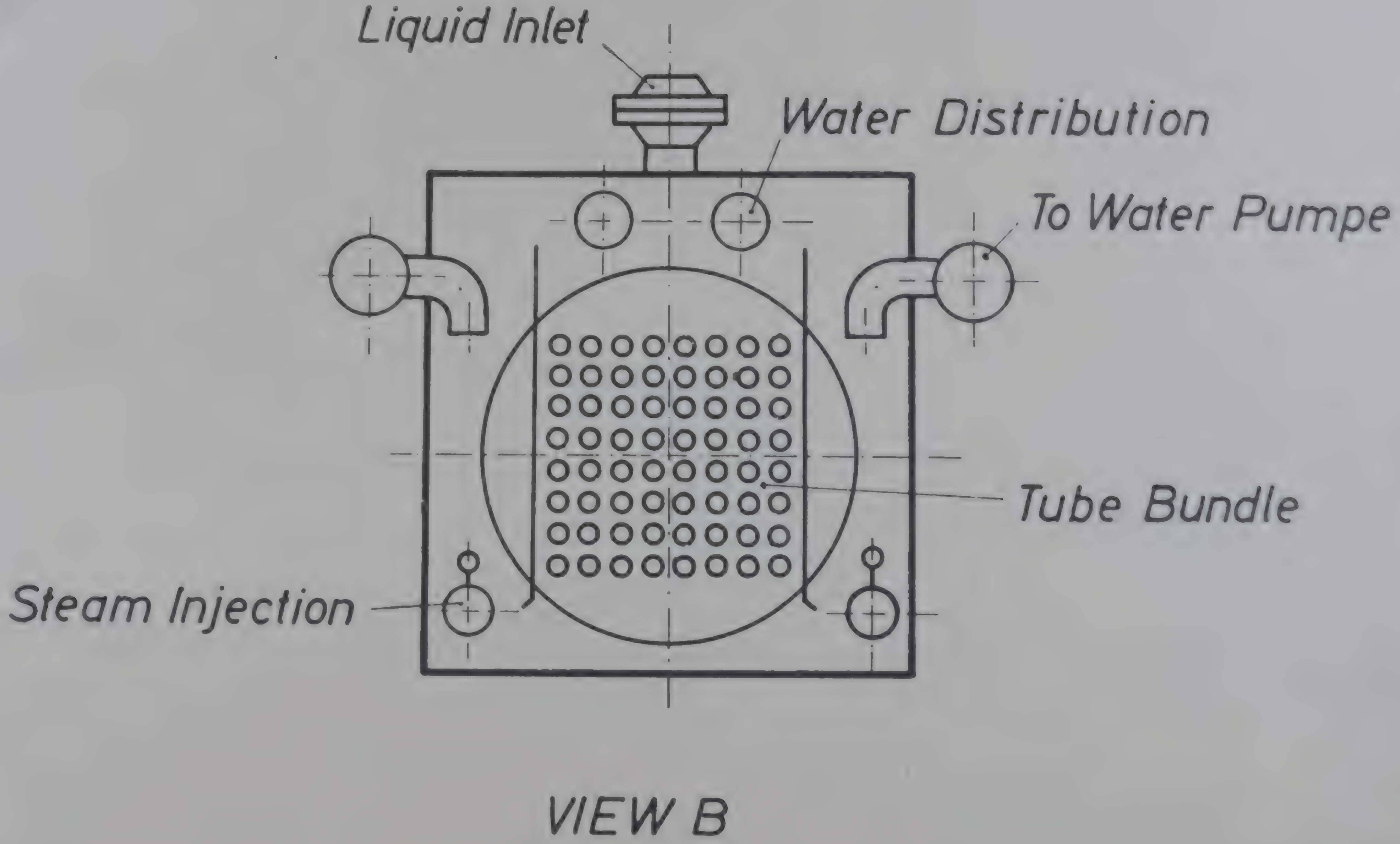

Liquid Inlet
Water Distribution
To Water Pumpe
Tube Bundle
Steam Injection
VIEW B

Fig.1b. Steam heated high performance evaporator

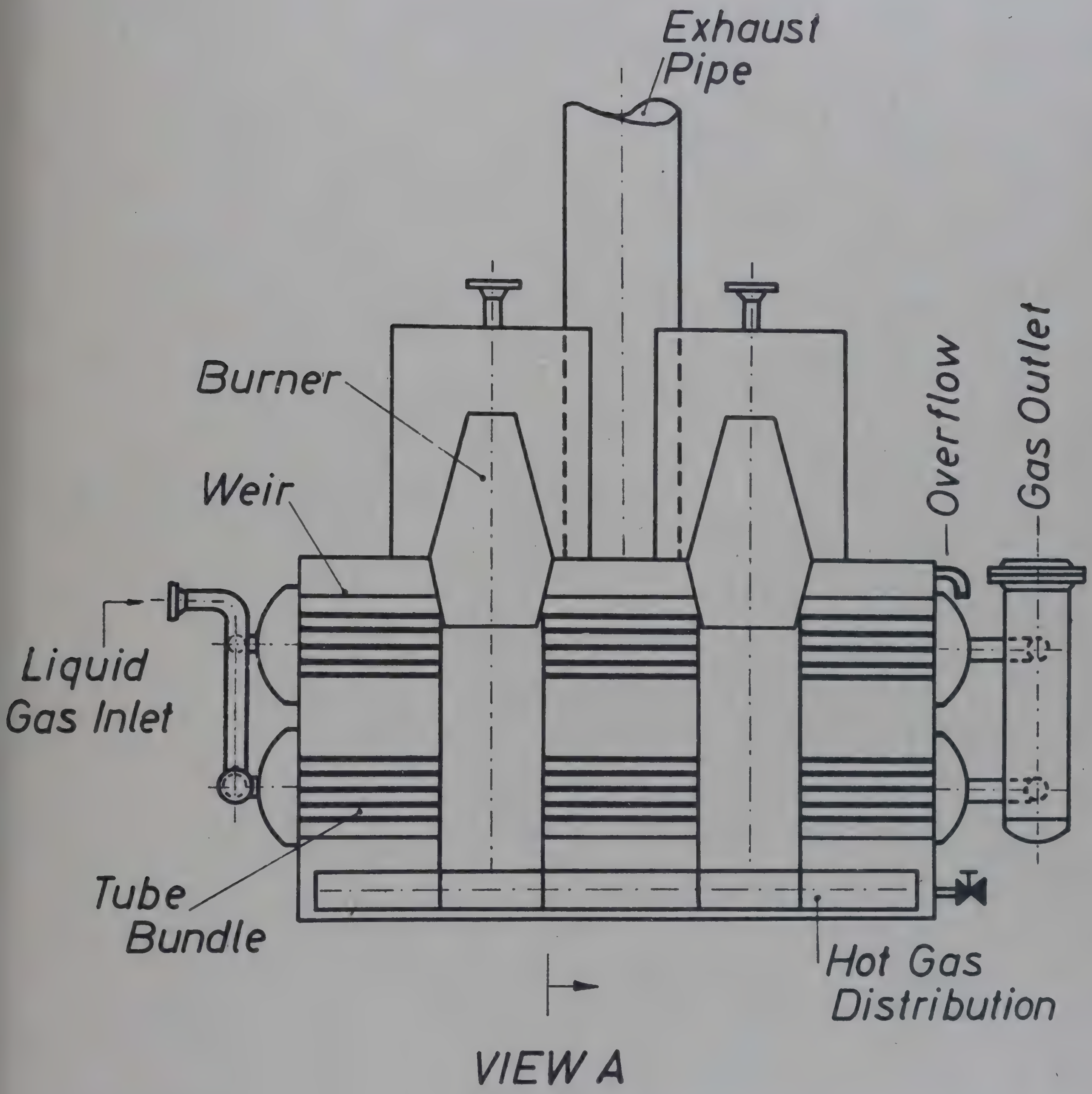

Exhaust Pipe
Burner
Weir
Liquid Gas Inlet
Tube Bundle
Overflow
Gas Outlet
Hot Gas Distribution
VIEW A

Fig.2a. High performance evaporator with submerged burner

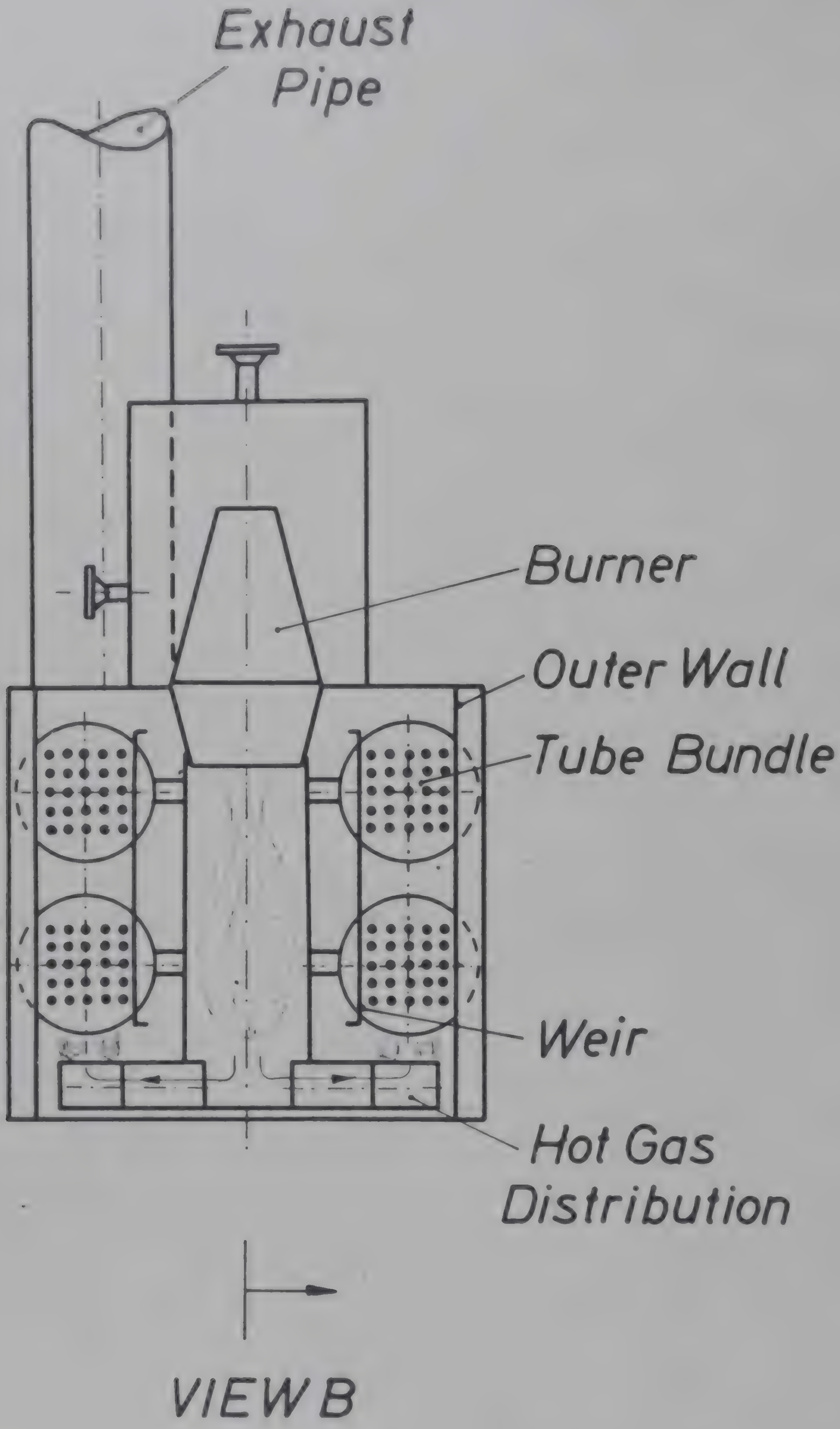
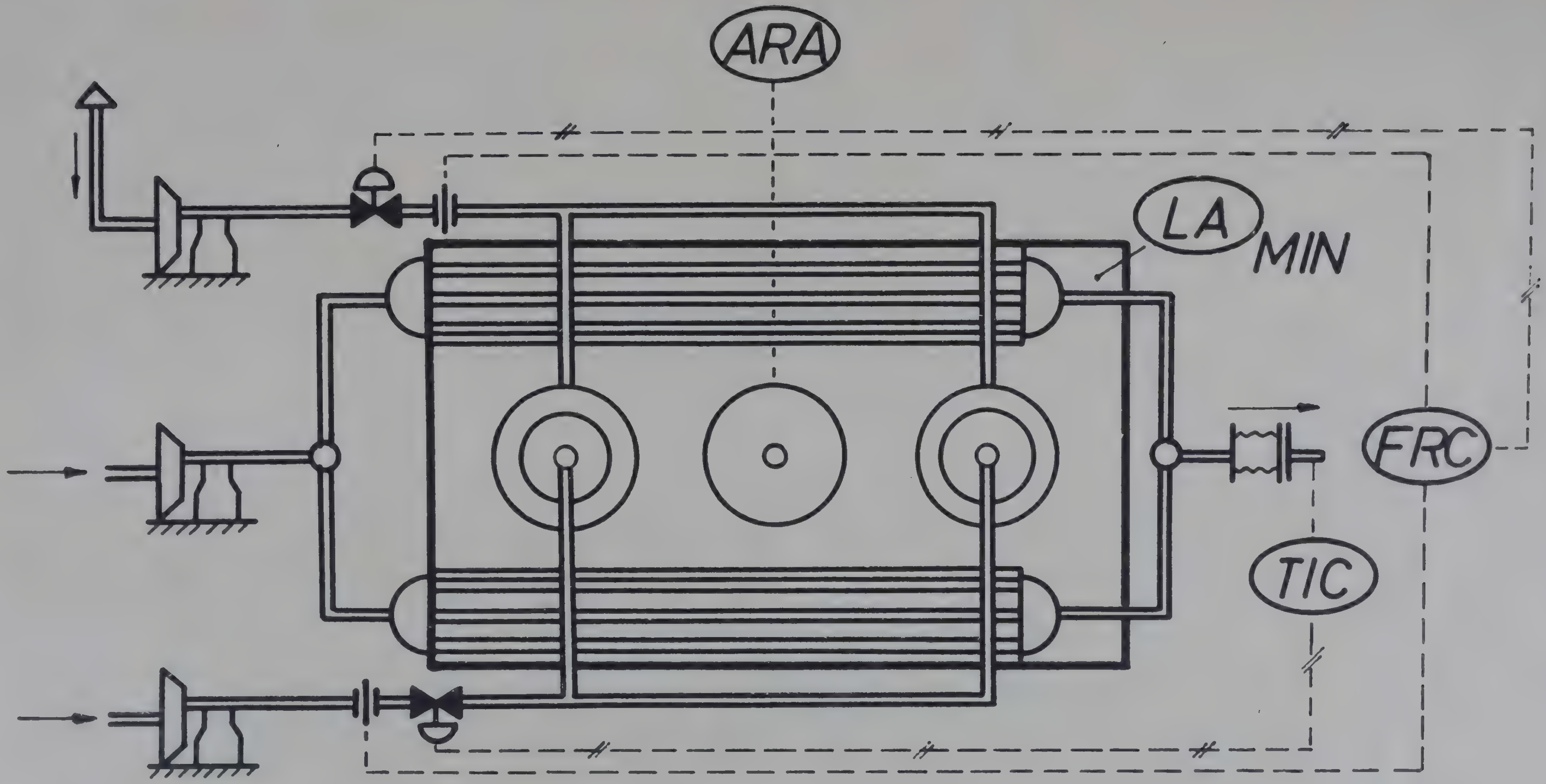
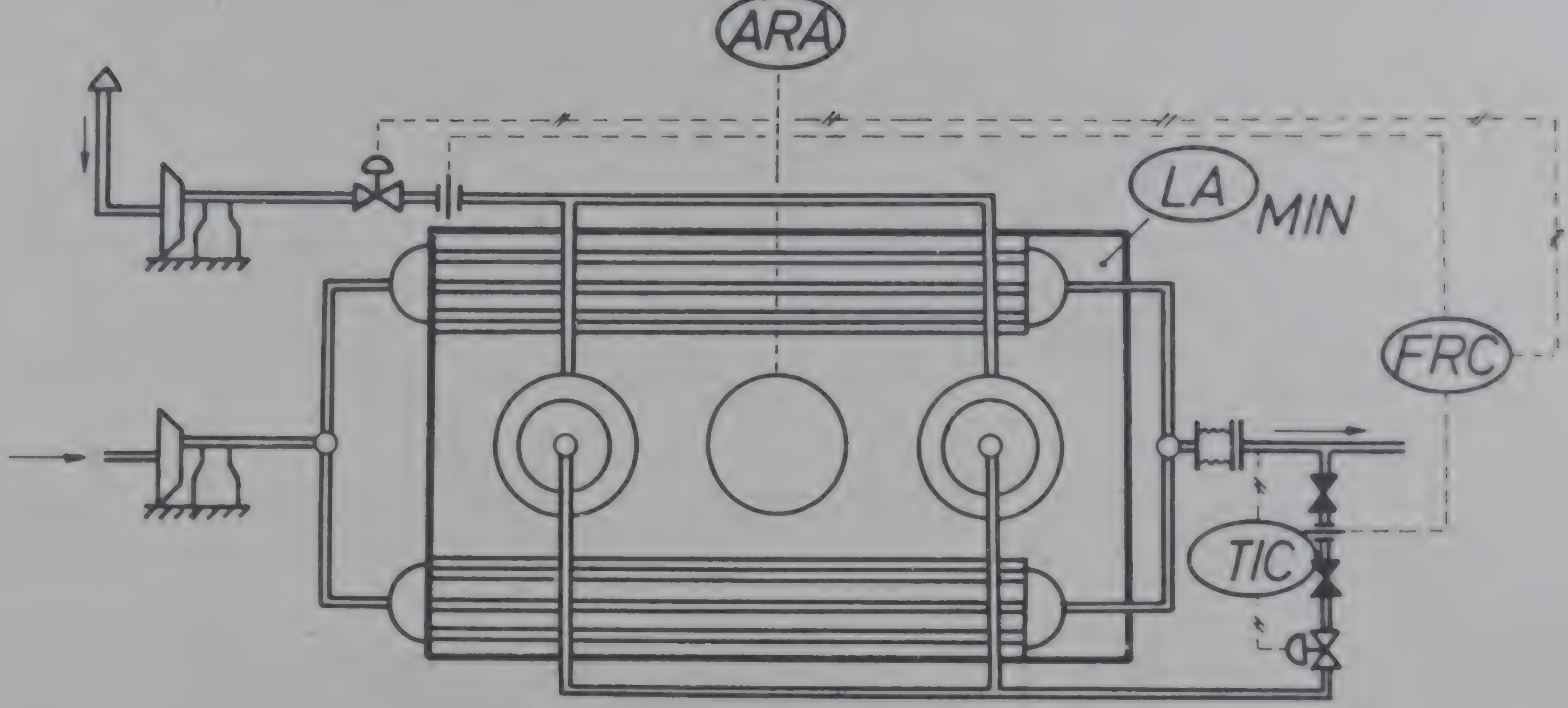

Exhaust Pipe
Burner
Outer Wall
Tube Bundle
Weir
Hot Gas Distribution
VIEW B

Fig.2b. High performance evaporator with submerged burner


ARA
LA
MIN
FRC
TIC

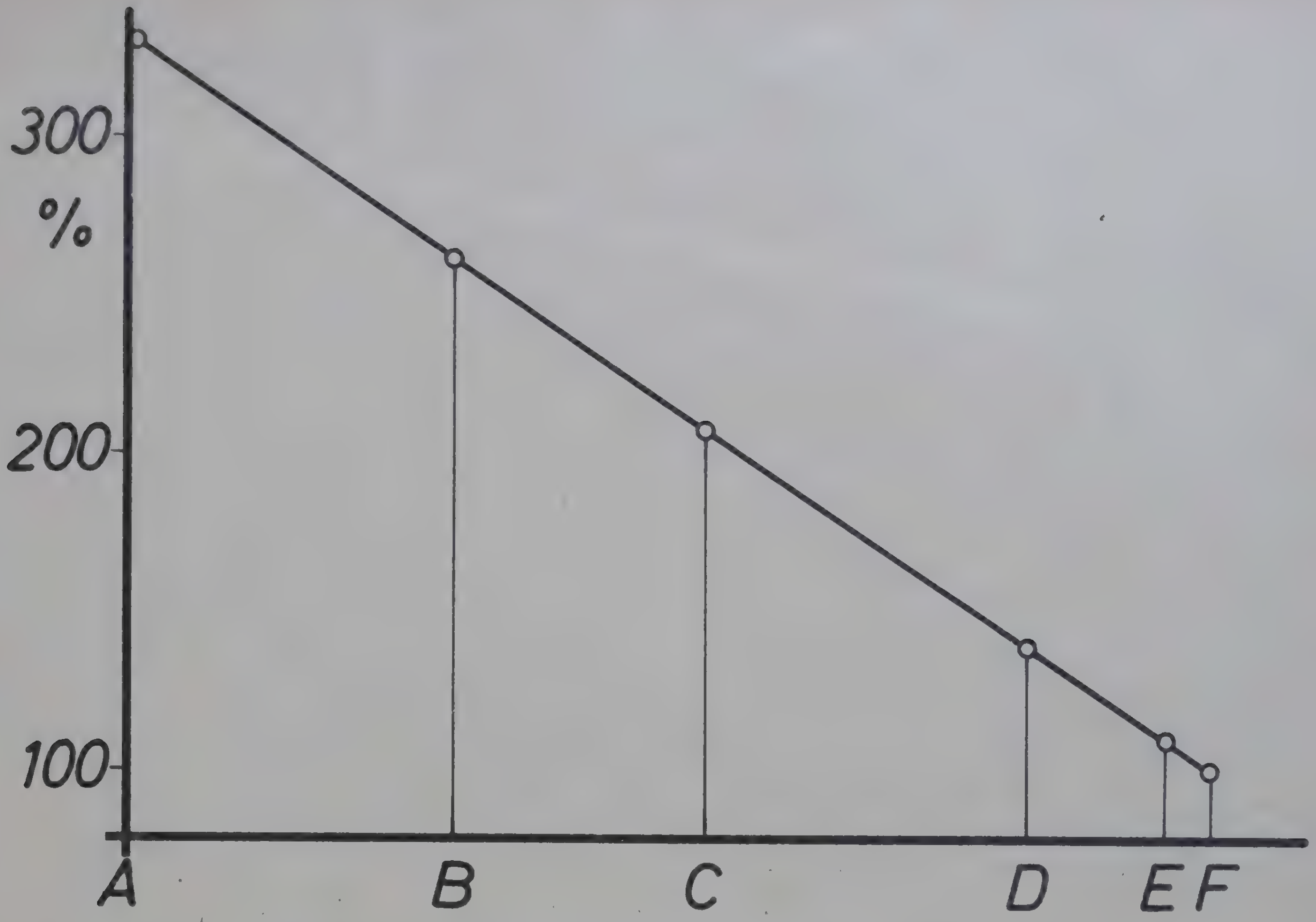
CASE I: USING PRESSING FREE FUEL FOR BURNER.

Fig.3. Assembled unit for revaporization of liquefied gas


ARA
LA
MIN
FRC
TIC

CASE II: USING GAS FROM PRODUCT STREAM EXPANDING IN BURNER.

Fig.4. Assembled unit for revaporization of liquefied gas



IN COMPARISON ARE THE FOLLOWING TYPES OF COMPLETELY PACKAGED EVAPORATORS.

Type		Material of tubes
A	Direct heated tubular furnace	Stainless steel
B	Water bath evaporator inclusive steam generator	Copper
C	High temperature oil cycle (1Burner)	Copper/stainless steel
D	Hot gas heated unit (1Combustion and dilution chamber)	Stainless steel
E	Hot gas heated unit (1Combustion and dilution chamber)	Alu
F	Submerged combustion technique with 3 burners.	Copper

Fig.5. Cost estimate for a vaporizer duty of 32 mm BTU/h

EXPERIENCE WITH FROZEN INGROUND STORAGE UNITS FOR LIQUEFIED NATURAL GAS - CANVEY ISLAND (U.K.)

By J. A. Ward* and P. C. Egan+

1. INTRODUCTION.

Between May 1968 and the present time three of four 21,000 ton frozen inground storage units at Canvey Island have been commissioned, and the fourth unit should be complete by July 1969. At the time of writing (November 1968) one unit is nearly full, another is half full, and it is expected that by the time maximum peak demands can be expected, in January and February 1969, two units will be full and a third unit half full. Thus some 2,500 mill. cu. ft. of peak load gas will be available which, if pumped out at a rate of 180 mm.scfd. to add to the normal 100 mm.scfd. base load, would last approximately 14 days.

During the time of the commissioning and early operation information has been collected on the amount of boil-off gas evolved, temperature movements in the soil and ice wall growth, ground heave and actual quantities of heat extracted from the soil.

2. HISTORICAL.

Following the inauguration of the scheme for the importation of L.N.G. into Canvey Island from Algeria, and with it the construction of the National Methane Grid, it was realised that additional storage capacity at Canvey would provide for increased flexibility of operation. This

* North Thames Gas Board
+ Gas Council

storage would enable liquid imported in the summer months to be stored against winter demand and would assist considerably in meeting winter peak loads.

The above-ground storage of 22,000 tons capacity existing in 1966 could not be considered as much more than buffer storage between shiploads bringing in 12,000 tons of L.N.G. every five days. Additional storage capacity was therefore urgently required, and it was estimated that storage of an extra 84,000 tons would meet the peak load demands for methane in this area for the winters of 1968/69, 1969/70 and 1970/71. This additional storage at the unloading terminal also fitted in to the Gas Council policy of having L.N.G. storage at the extremities of the pipeline system in the country, since Canvey Island is at the south-eastern end of the main backbone grid pipeline which will be extended into Scotland by 1970. The recently announced L.N.G. storage and liquefaction facility at Glenmavis, Scotland, will form a complementary L.N.G. store at the northern end of the grid pipeline. An aerial view of the inground units, constructed by Sir R.McAlpine & Sons Ltd. as main Contractor, is shown in Figure 1.

A number of factors influenced the choice of frozen inground units. At the size required of 300,000 bbl. capacity (21,000 tons) they were competitive in price with surface tanks including foundations. In addition they are unobtrusive to nearby habitation, safe from fire and liquid spillage and they take up less ground space compared with above-ground storage as bunds are not required. The 16,000 ton capacity frozen inground storage unit at Arzew in Algeria had gone to work successfully, and the difficulties which were to be encountered with inground installations in underlying rock formations were not known at this time. The ground strata on the Canvey Island site, consisting mainly of silt, gravel and clay containing up to 30% of moisture, appeared quite suitable for L.N.G. storage and for the freezing techniques adopted for the excavation.

3. NOTES ON THE INSTALLATION.

Apart from the common use of sections of the fill lines from the jetty, the inground storage units form part of a completely separate storage, pumping and evaporating facility capable of delivering 180 mm.scfd. of gas into the grid main. Cross connections are provided between the inground installation and the above-ground system which allow the spray cooling and the filling of the inground units to be done from the above-ground tanks, and L.N.G. at 1000 p.s.i.g. can be pumped from the inground installation to both the direct-fired peak load evaporators or the base load water heated evaporators, or, alternatively, from the above-ground installation to both types of evaporators.

The design and construction of these inground units has been described in papers to the First International L.N.G

Conference at Chicago and to an I.I.F. Conference in Madrid, (refs. 1 and 2) so only a brief outline is given here. The four inground units, each of 21,000 tons capacity, are 130 ft. in diameter and 130 ft. deep. Prior to excavation, the freezing technique consisted essentially of forming a frozen base plug and a frozen cylindrical wall of sufficient strength to withstand the hydrostatic forces when the ground was excavated from inside this ice cup. Cold brine circulated through some 41 tubes in the plug, and through two concentric rings of 90 tubes each in the walls, formed the required ice structure. A cross section of a unit depicting soil strata, thermocouple points and outline details is shown in Figure 2.

The rate at which such an ice vessel can be formed in the ground is controlled by the driving force for heat transfer from the ground to the cold interface, and from there to the refrigerant brine, and to the water content in the soil. The driving force is dependent upon brine and soil temperatures and upon the thickness of ice around the freeze tubes.

At the outset of freezing heat transfer rates are high as the interface is virtually the cooling tube surface, and is at the actual brine temperature. With the growth of ice the rate increases, since the area for heat transfer increases. At some point the thermal resistance of the ice begins to control heat flow and the effective rate of increase in heat transfer areas is reduced as the ice cylinders formed around adjacent tubes join together. Therefore as the ice wall thickens further the "pull-down" rate decreases.

This is illustrated by Figure 3, where it is seen that maximum "pull-down" is effected some 12 weeks after commencement of freezing. Thereafter progress is dependent upon the maintenance of low brine temperatures.

Initially refrigeration was derived from mechanically driven ammonia compressing plant, from which brine at -20°F. was obtained. Later a plant was commissioned which enabled the extraction of cold energy from L.N.G. to the brine via an intermediate propane circuit (ref. 3). Brine temperatures of as low as -25°F. are possible from this plant.

A heavy concrete ring beam cast originally in twelve separate sections with 4 ft. gaps which were concreted in when frost heave became uniform, is positioned round the lip of each unit and supports the aluminium roof. The framework of the roof supports a secondary structure on which is laid expanded ebonite insulation and this insulation is continued down the inside face and underneath the ring beam.

Each unit has two submerged Carter pumps, both capable of pumping 150 tons/hour at 120 p.s.i.g. through 6 ins. diameter aluminium pipelines to the booster pumps. The four booster pumps, each capable of handling 50 tons/hour, raise the pressure to 1000 p.s.i.g. at the inlet to the three direct-fired evaporators. These evaporators have Sub X type methane burners which maintain a water bath at 130°F. The

L.N.G. is vaporised in a stainless steel bundle submerged in the water bath. The layout is illustrated in Figure 4.

Boil-off gas from the four units enters a common 30 ins. diameter pipeline where the flow rate is measured by a Venturi meter, after which the boil-off gas passes through a steam heater to raise the temperature to 0°C. before entry to the boil-off gas compressors.

The boil-off connection to each unit contains a butterfly valve which is controlled automatically to maintain the internal pressure at a fixed absolute value to prevent boil-off surges that otherwise arise from rapid barometric changes.

4. COMMISSIONING.

(a) Pre-Cool Down.

With this type of storage unit already at -10°C. inside prior to cool-down, a thorough inspection is required to insure that all equipment is ice-free, e.g. ice on the tape gauge guide wires which would prevent free movement of the float, ice in the pump suction pots and frozen pump foot valves. Once the pumps have been installed it is necessary to keep them in an atmosphere of nitrogen to prevent ingress of moisture into the motor windings. All loose ice and other debris must be cleared from the floor and shoulders.

After pressure and vacuum testing nitrogen purging was carried out using vaporised nitrogen from liquid nitrogen tankers. The nitrogen was admitted through the boil-off outlet and a relief valve fitting in the top of the roof crown, the air being forced out from the bottom up through the fill pipe to atmosphere. The relatively warm nitrogen pushing down on the cold denser air had a piston effect which kept mixing to a minimum, and each unit required only 3 mill. cu. ft. of nitrogen for a container volume of some 2 mill. cu. ft., in order to reduce the oxygen content at all levels below the 7% required for safe gassing up.

Gassing up was done in a similar manner, methane gas being admitted via the boil-off outlet, and nitrogen ejected up through the fill pipe.

At this stage the unit concerned was ready for cool-down to begin.

(b) Cool-down.

The mode of cool-down was agreed in all respects with Conch Methane Services Ltd. prior to commencement, and essentially this was to cool the vessel slowly and uniformly in order to avoid excessive temperature gradients and thermal shock. This was achieved by injecting L.N.G. at controlled rates through the centre spray, such that an internal circulation of cold gas was set up down the centre and upwards past the walls.

The spray rates were to be controlled rising from 2 tons to 6 tons/hour of L.N.G. such that the fall in temperature, as measured by the temperature of the boil-off gas, amounted to some 4½°C./day.

This process is illustrated by Figure 5. Actual cool-down achieved is shown by curve 1A, and it is seen that this curve flattens out during June at approximately -105°C. This point represents the limit of the cooling ability by the spray coupled with a buffering effect created by the presence of heavy ends which have been collecting on the bottom. The spraying technique was proving effective, as the wall temperatures were falling very uniformly at various levels in the unit and also kept close to the actual boil-off gas exit temperature.

The addition of L.N.G. direct on to the base via the fill line - shown by curve 3 - enabled temperatures to continue downwards as shown. Curve 2A shows actual boil-off which is fully controllable up to the point of liquid accumulation on the base.

(c) First Fill.

Sub-cooling of the ice vessel continues during the first fill with cold liquid, the amount of heat removed being very considerable. Consequently the rate of fill has to be controlled just as the spray is in the earlier phase, each addition of liquid giving rise to a temporary surge in boil-off.

It was in fact possible to deliver enough liquid to the base so that the level rose at some 2 ins./hour (660 tons /day), without the boil-off rate exceeding some 8 to 9mm.scfd.

During this period of some 14 days filling the spraying rate was progressively reduced and was finally stopped when the liquid level had reached 40 ft. The exit vapour temperature had been maintained at between -90°C.and -100°C. during this early filling period with combined bottom filling and spraying, and when the spraying was stopped the vapour temperature rose to and remained at -50°C. A careful watch was kept on temperatures in the ice wall near the shoulder of the unit, as it was vital that ice wall growth should continue in this area under the ring beam. From -77°C. the shoulders rose in temperature to -49°C over a period of 26 days, and then gradually began to fall again as the liquid level was further increased.

The graph in Figure 6 illstrates the pattern of boil-off gas rates obtained during the initial cool-down and first filling phase. The boil-off rate climbed steadily as the spraying rate was increased to keep the cool-down rate constant, and then flattened out as the spray rate reached a maximum and heavy ends began to collect on the floor. A sudden increase occurred as bottom entry of L.N.G. was commenced with the spraying maintained. A further flattening out resulted as the spray rate was cut back to keep the total boil-off rate down to around 7 mm.scfd., followed by a further surge upward when the temperature on the bottom of the unit reached -160°C. and filling of the unit began. A peak boil-off rate of 8.8 mm.scfd. was reached when the liquid level had risen to 40 ft. and at this stage spraying was stopped. The cessation of spraying resulted in a rapid fall in the boil-off rate to 3.6 mm.scfd. or 1.13% of the tank contents per day at the 40 ft. level.

As the initial filling continues the boil-off rate at successively higher levels is shown with the corresponding boil-off as a percentage of the tank contents. The falling curves from the initial fill line indicate the fall off in boil-off rate with time at given levels. For example by the 8th November 182 days after the start of cool-down, the boil off at the 108 ft. level had dropped to 1.0% of tank contents per day, compared to 1.28% when the level had first reached 108 ft. 144 days after the start.

These boil-off rates should continue to fall at a decreasing rate over a period of some 2 years, due to the increasing thermal resistance of the growing ice wall, when it is expected that a state of equilibrium will be reached.

3. QUANTITIES OF HEAT INVOLVED DURING THE COOLDOWN.

The heat removed during the cooldown can be estimated from the quantity of L.N.G. that is evaporated. It comprises the sensible heat actually removed from the ice vessel and the heat inleak into the vessel during the period of time in question.

Although this has been the subject of detailed analysis, only an indication of the quantities involved is given here as follows:-

Assume the walls and base of the vessel are 20 ft. thick at the commencement of cooldown:

based upon derived data)

* the densities of frozen silt and frozen clay are 101 lbs./cu.ft. and 115 lbs./cu.ft. respectively;

* specific heats are 0.31 and 0.244 respectively; average temperature of walls before and after cooling are -3°F. and -114°F. respectively; average temperature of base before and after cooling are -18°F. and -109°F. respectively.

Then: heat removed from walls (silt)	≡ 4080	tons	LNG	vaporis
" " " " (clay)	≡ 3140	"	"	"
" " " base	≡ 2070	"	"	"
Total equivalent LNG vaporised	≡ 9290	"	"	"
Total measured boil-off	≡ 25000	"	"	"
Boil-off arising from sub-cooling	≡ 9290	"	"	"
Difference due to heat in-leakage	≡ 15710	"	"	"

Over a period of 156 days for spraying and filling, the average boil-off due to heat inleakage $= \frac{15710}{156} = 101$ tons/day

Allowing for the change in driving force for the cold vessel, the heat inleakage after cooldown will be approximately three times the inleakage before, and taking an average of 101 tons/day of liquid evaporated, if the daily evaporated rate at the end of cooldown = R.

$$\text{then } \frac{R + \frac{1}{3}R}{2} = 101$$

$$\text{i.e. } R = 152 \text{ tons/day}$$

Equivalent boil-off rate for a full tank after cooldown

$$= \frac{152 \times 100}{20{,}000} = \underline{\underline{0.76\%}}$$ per day at the point when the unit is full for the first time.

It can be seen that there is a reasonable measure of agreement between this figure and the values given in Figure 6, taking into account the fact that no allowance has been made for heat inleakage through the roof.

6. RING BEAM MOVEMENT AND TEMPERATURE GRADIENTS DUE TO SUB-COOLING.

After the ring beam sections had been cast a considerable amount of settlement took place over a three month period, a typical section dropping some 5 ins. on the inner side and nearly 4 ins. on the outside edge. The plug freeze had been under way for 8 weeks prior to casting, and wall freeze commenced 6 weeks after casting. Settlements of this degree had not been observed in any area of the site away from the inground storage, and it is possible that water migration towards the plug left the soil above unsaturated and hence more readily compacted by the weight of the beam sections. The inside faces of the sections continued to fall after the outside faces had stopped causing the beams to rotate slightly inwards.

Uniform heave then took place over a period of six months until the outside edge became stationery, but the inner side continued to rise slowly, tending to cause an outward rotation of the beam. Since all the 12 sections had been concreted together when the upward heave became uniform, this had the effect of causing cracks to occur in the top outer face of the ring beam. The highest level reached by the outside edge was some 2.25 ins. above the original casting level, with a corresponding measurement for the inside edge of 1.7 ins. Thus, total heave was of the order of 6.5 ins.

During the spray cooling and filling operation the ring beam had dropped slightly in a uniform manner which indicated that the frozen soil had shrunk with a fall in temperature to -100°C. in the body of the ice 11 ft. from the face of the excavation. It is possible that as the ice plug in the base of the unit increases in depth with time, further heave of the ring beam may take place.

With the spread of the ice wall beyond 30 ft. from the periphery of the unit, some 2 ins. of settlement has already occurred on pipeline trestle bases carrying pipes away from the units in a manner similar to the ring beam movement before heave started. Regular measurements of the levels of the units and these supports are taken to ensure that the relative movement does not exceed that allowed by the bellows units in the pipelines. If necessary, support hangers are adjusted.

Temperature gradients in the ground strata are of interest as the liquid level rises and the graph in Figure 7 shows these gradients at three different distances from the inside face of the wall at 60 ft. to 70 ft. depth. As the liquid level is approaching the level of these thermocouples a temperature drop of 5°C./day occurs 3 ft. back from the face, and 1.4°C./day 11 ft. back. 29 ft. back from the wall the effect at this time is imperceptible, but by the time the liquid level had reached 70 ft. the gradient here became 1°C./day.

7. OPERATIONAL CONTROL.

L.N.G. of a given composition gives rise to boil-off gas having a larger proportion of lighter constituents, hence as boil-off proceeds calorific value of the residual liquid slowly rises.

Since, as indicated above, the boil-off rate from frozen ground stores tends to be high, this effect is accelerated and special procedures are necessary to ensure that the calorific value of the stored liquid is kept within desired limits. This is achieved by switching stocks, i.e. rich liquid is periodically exchanged for a batch of the normal run of fresh liquid being evaporated from the Station.

In order that the evaporated Algerian Gas will be compatible with North Sea Gas it is diluted with lean reformed gas, supplied from two I.C.I. type continuous catalytic reformers to give a Wobbe No. in the final gas of 1335. The lean gas is compressed to 1000 p.s.i.g. by reciprocating compressors prior to injection into the evaporated methane. Automatic Wobbe No. control is maintained on the final gas mixture which regulates the amount of lean gas injected, and safety trips shut off the dilution gas under low methane flow and Wobbe No. conditions.

8. CONCLUSIONS.

The units discussed have been constructed at Canvey and successfully commissioned. Whilst the boil-off rate has not been as low initially from these units as anticipated, there are signs that the boil-off rate will decrease to reasonable proportions in time. Although compression costs are involved, this boil-off gas is readily absorbed in the local distribution system and forms part of the outgoing base load.

The large quantity of L.N.G. stored, coupled with the high rates of evaporation possible at high pressure, make a valuable contribution in meeting the peak load and emergency requirements of the national grid pipeline system.

Acknowledgments

The Authors would like to thank their colleagues for their help, and the North Thames Gas Board and Gas Council for permission to publish this Paper.

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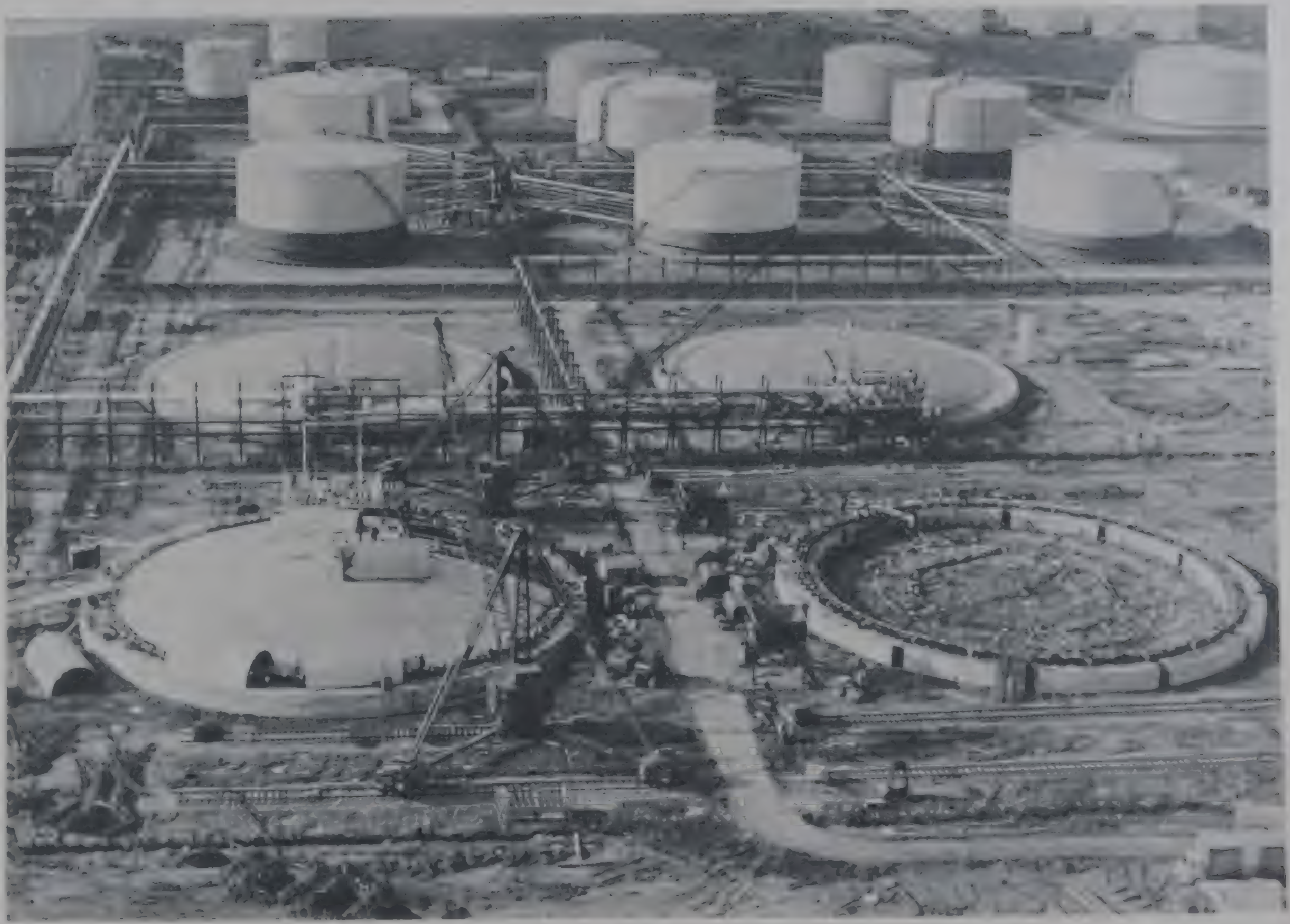


Figure 1. Aerial view of frozen inground units.

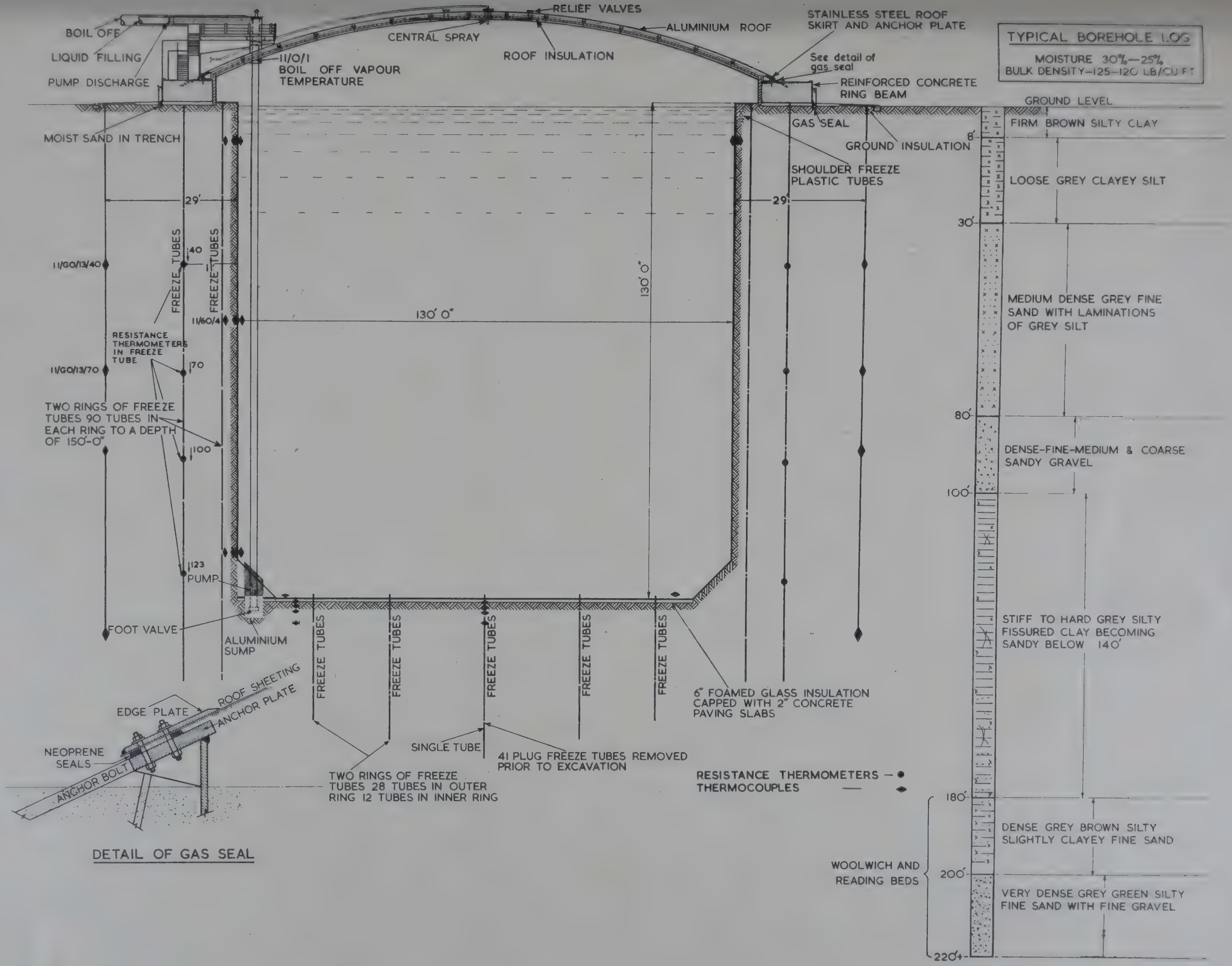

BOIL OFF
LIQUID FILLING
PUMP DISCHARGE
RELIEF VALVES
CENTRAL SPRAY
ROOF INSULATION
ALUMINIUM ROOF
11/O/1
BOIL OFF VAPOUR TEMPERATURE
STAINLESS STEEL ROOF SKIRT AND ANCHOR PLATE
See detail of gas seal
REINFORCED CONCRETE RING BEAM
TYPICAL BOREHOLE LOG
MOISTURE 30%—25%
BULK DENSITY—125—120 LB/CU.FT
GROUND LEVEL
MOIST SAND IN TRENCH
GAS SEAL
GROUND INSULATION
SHOULDER FREEZE PLASTIC TUBES
29'
29'
130' 0"
130' 0"
FREEZE TUBES
FREEZE TUBES
40
11/GO/13/40
11/60/4
RESISTANCE THERMOMETERS IN FREEZE TUBE
70
11/GO/13/70
TWO RINGS OF FREEZE TUBES 90 TUBES IN EACH RING TO A DEPTH OF 150'-0"
100
123
PUMP
FOOT VALVE
ALUMINIUM SUMP
FREEZE TUBES
FREEZE TUBES
FREEZE TUBES
FREEZE TUBES
FREEZE TUBES
6" FOAMED GLASS INSULATION CAPPED WITH 2" CONCRETE PAVING SLABS
SINGLE TUBE
41 PLUG FREEZE TUBES REMOVED PRIOR TO EXCAVATION
TWO RINGS OF FREEZE TUBES 28 TUBES IN OUTER RING 12 TUBES IN INNER RING
RESISTANCE THERMOMETERS —●
THERMOCOUPLES —◆
EDGE PLATE
ROOF SHEETING
ANCHOR PLATE
NEOPRENE SEALS
ANCHOR BOLT
DETAIL OF GAS SEAL
FIRM BROWN SILTY CLAY
8'
LOOSE GREY CLAYEY SILT
30'
MEDIUM DENSE GREY FINE SAND WITH LAMINATIONS OF GREY SILT
80'
DENSE-FINE-MEDIUM & COARSE SANDY GRAVEL
100'
STIFF TO HARD GREY SILTY FISSURED CLAY BECOMING SANDY BELOW 140'
180'
DENSE GREY BROWN SILTY SLIGHTLY CLAYEY FINE SAND
200'
VERY DENSE GREY GREEN SILTY FINE SAND WITH FINE GRAVEL
220'+
WOOLWICH AND READING BEDS

Figure 2. Typical cross section of a unit showing thermocouple points, soil strata, etc.

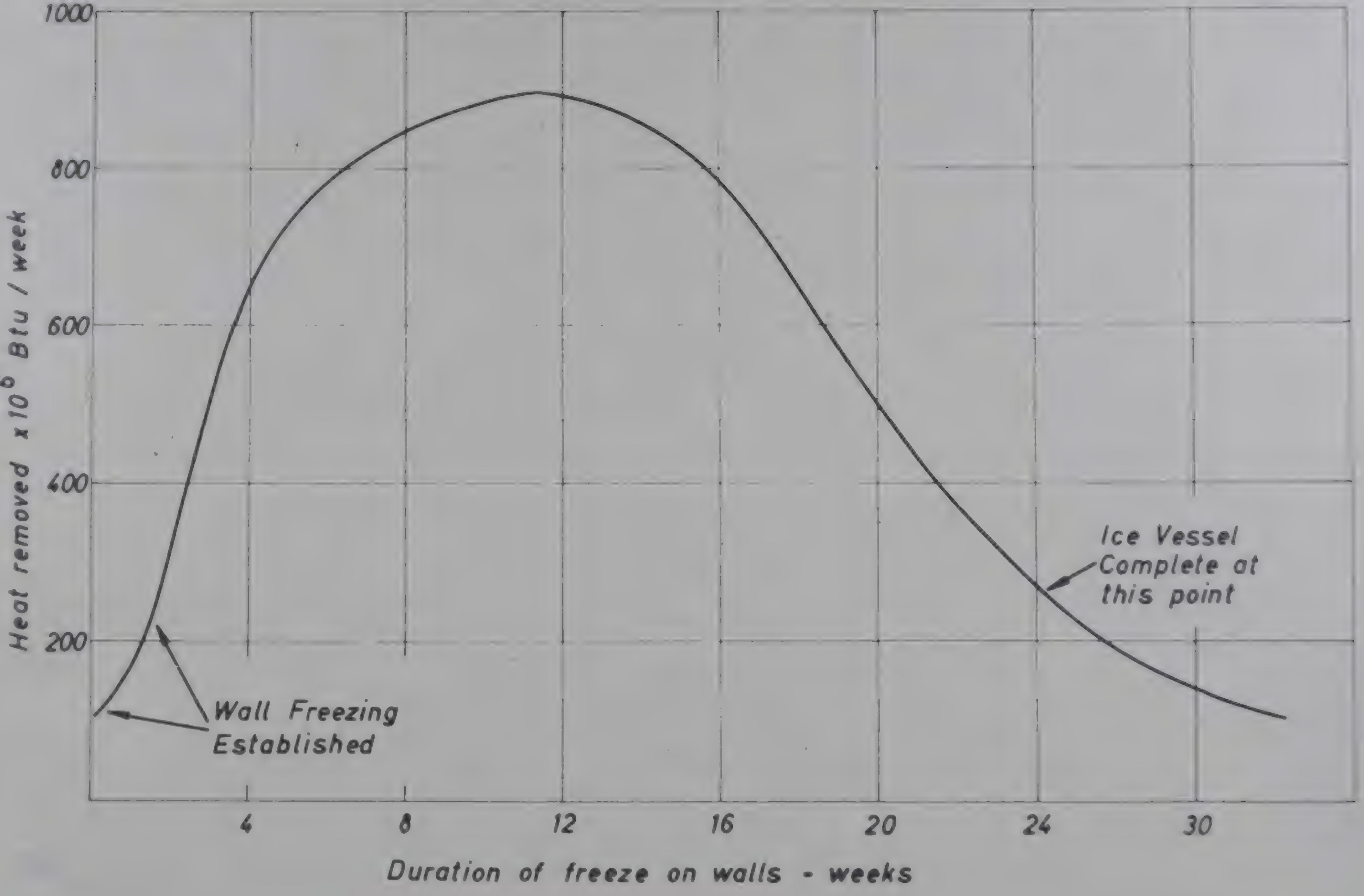


Figure 3. Ground freezing curve.

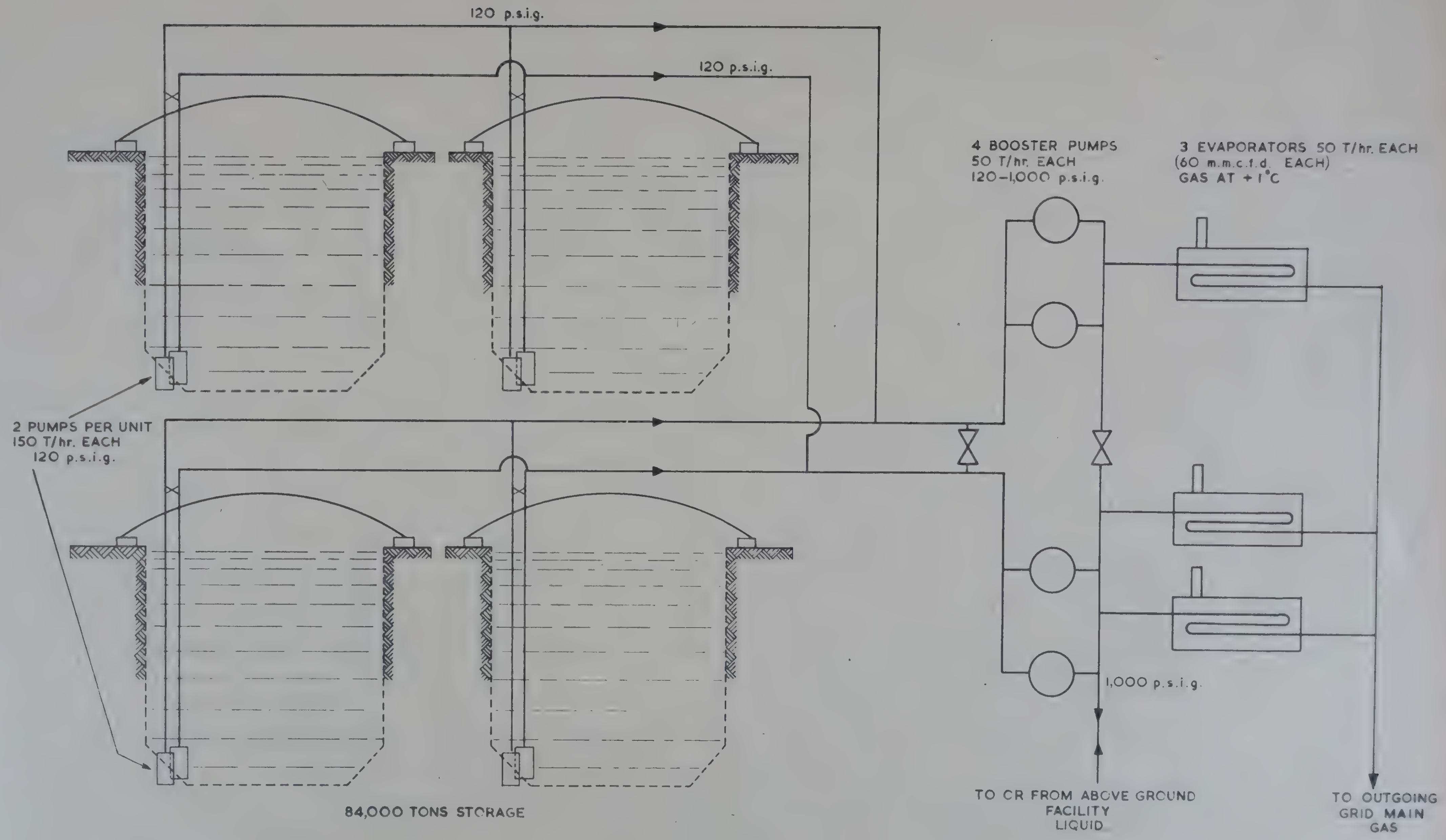

120 p.s.i.g.
120 p.s.i.g.
4 BOOSTER PUMPS
50 T/hr. EACH
120–1,000 p.s.i.g.
3 EVAPORATORS 50 T/hr. EACH
(60 m.m.c.f.d. EACH)
GAS AT + 1°C
2 PUMPS PER UNIT
150 T/hr. EACH
120 p.s.i.g.
1,000 p.s.i.g.
84,000 TONS STORAGE
TO CR FROM ABOVE GROUND
FACILITY
LIQUID
TO OUTGOING
GRID MAIN
GAS

Figure 4. Line diagram of L.N.G. layout.

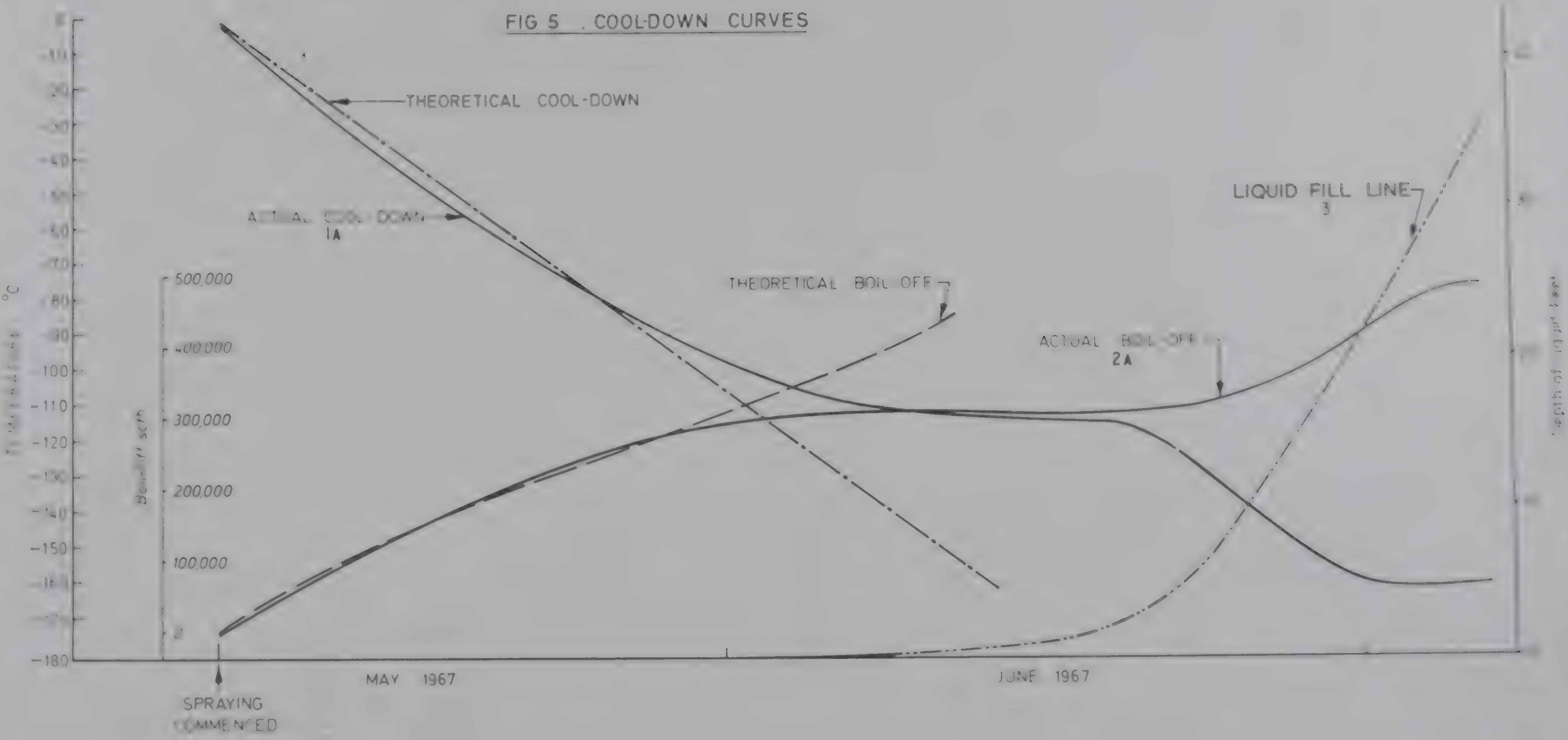


Figure 5. Cooldown curves.

Fig 6 No. 11 UNIT
BOIL OFF RELATED TO COOLDOWN
INITIAL FILLING AND ICE WALL GROWTH

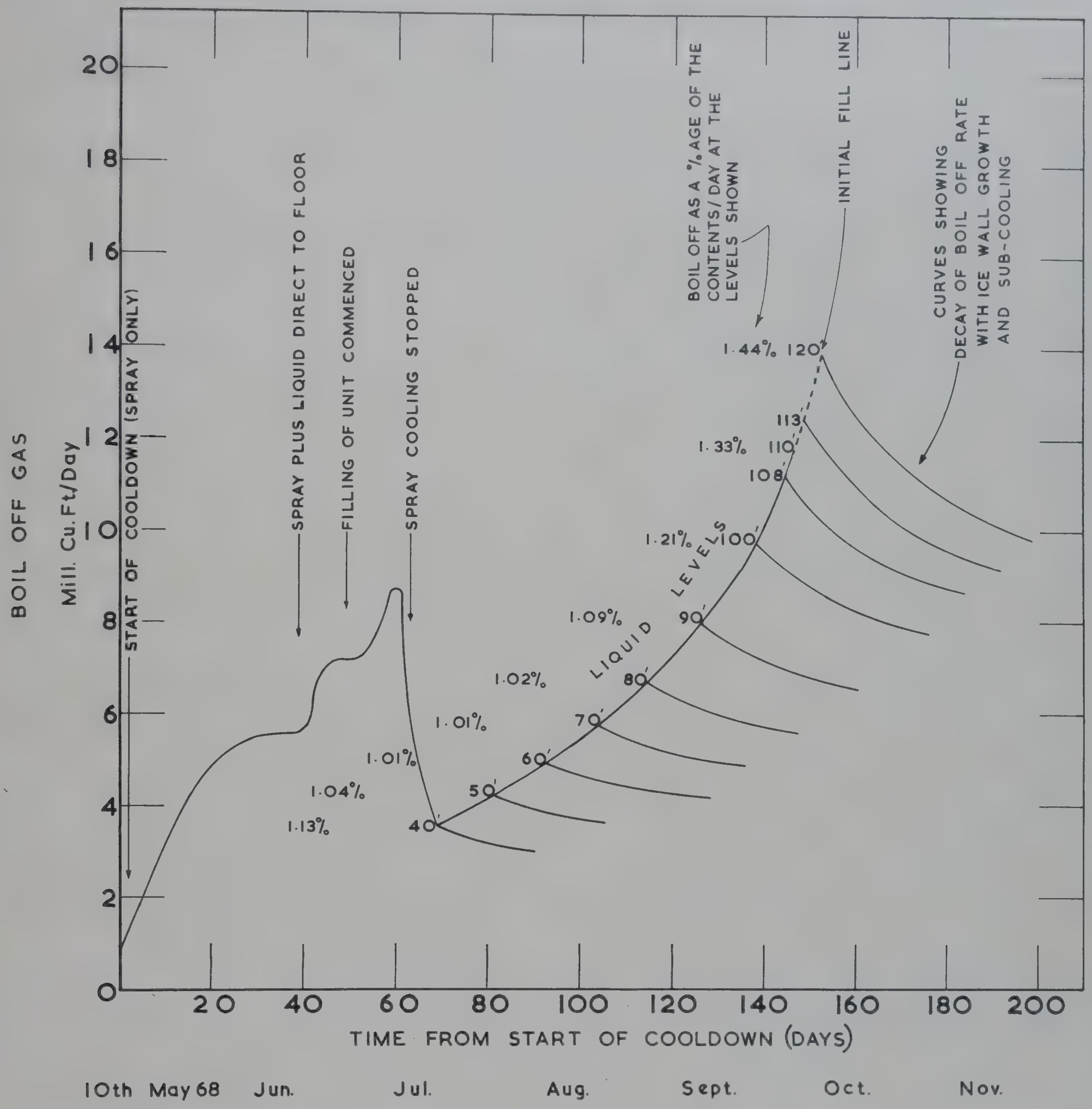


Figure 6. Boil-off gas rate curves related to cooldown, filling and ice growth periods.

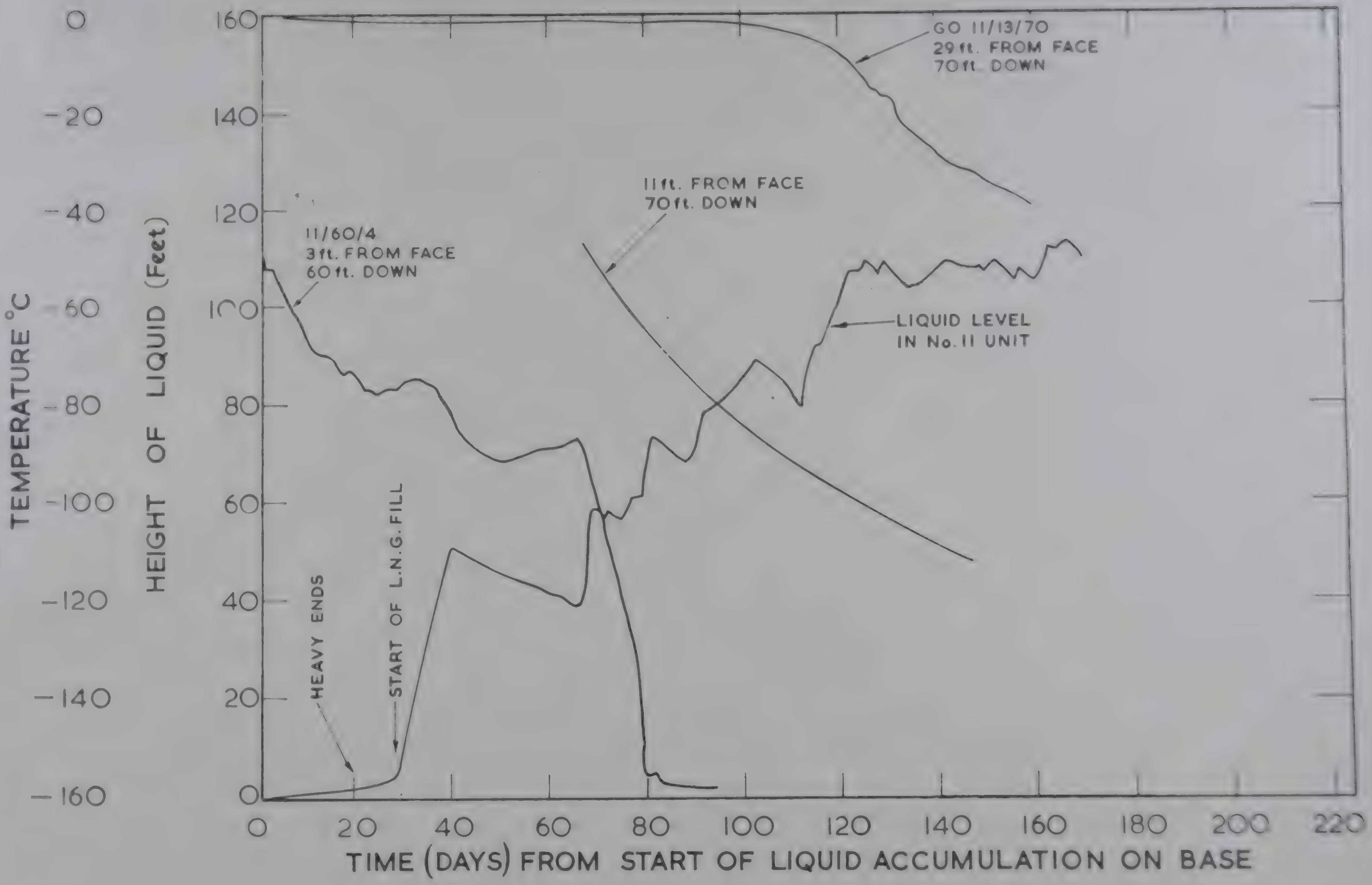


Figure 7. Curves showing change in ground temperature and liquid level with time.

"Design and Construction of Roofs Over Inground Reservoirs"

W.Hamilton, M.I.C.E.   G.P.Manning, M.Eng., F.I.C.E., M.Cons.E.

DESIGN OF ROOFS - GENERAL

The natural shape for an inground reservoir is circular in plan, the wall of frozen ground round the reservoir acting as a ring in simple compression to resist the lateral pressure from the surrounding ground and groundwater.

The great depth of the reservoir, high ground heave andthermal movements exclude the idea of any central support. These conditions indicate the adoption of some type of dome for the roof.

Again the simplest way of containing an appreciable gas pressure over a circular reservoir is by a spherical dome in simple tension. This also indicates the provision of a domed roof.

The Hamman spherical dome, of which some 40 examples have now been built, has been used for the roofs at Arzew and Canvey.

For the sake of economy the foundations that support or hold down the roof are placed as near as possible to the hole. This means that they stand on ground that will eventually be frozen and they will certainly move upwards and may move laterally. A uniform upward movement is easy to cope with. At Port Arzew a general rise of 15" has occurred while at Canvey a final figure of 24" has been calculated.

Irregular upward movement and lateral movement(whether regular or irregular) are more difficult to deal with.

Before deciding on what type of roof to adopt it is necessary to make a study of ground conditions down to a depth of 50'0" below the bottom of the reservoir and make an estimate of the ground movements that will occur during construction and in service. If ground freezing is used as a method of construction the time schedule of the freezing programme must be studied.

If large and irregular movements are expected they may be dealt with by constructing a deep circular retaining wall (or caisson) capable of resisting or absorbing radial movement and capable of riding over the crests and troughs formed in the ground by irregular vertical movement.

Alternatively small separate foundations may be provided and a type of roof designed that has sufficient means of adjustment and sufficient flexibility to absorb these movements.

On a really bad patchy site the designer must think in terms of sliding bearings, pin joints, long flexible suspenders and three-point-supports with adjusting wedges or screws everywhere.

The main contractor's construction methods and construction programme must be studied. This may be such that the roof cannot be assembled in its final position but must be assembled on an adjoining site then lifted or rolled into place.

In most cases large openings in the roof must be left open until the last minute for the main contractor's access.

Loading

The upward pressure of the gas must eventually be offset by some form of Kentledge. This may be concrete or earth ballast dry or wet and possibly frozen. Earth ballast must not be used in any position where it might be removed by some uninformed person. This Kentledge may be spread uniformly over the roof (as at Port Arzew) in which case the roof carries a heavy downward load under vacuum conditions or it may be spread round the outside of the roof (as at Canvey) in which case the roof carries a leavy upward load under maximum gas pressure. In some cases a compromise between these two arrangements may be best.

Insulation

The roof consists essentially of three parts:-

1. A supporting structure
2. A gas-tight membrane
3. A layer of insulation

not necessarily in this order as the insulation may be above or below the gas-tight membrane and above or below the supporting structure.

All members below the insulation must be of material capable of functioning at low temperatures and all construction below the insulation must be designed to cope with thermal contraction down to -160 deg.C. This is usually a most cogent argument for putting the insulation at the bottom of the sequence. At Port Arzew and Canvey the insulation is carried on a light framework close below the gas-tight membrane. It would have been more convenient, but more expensive, at Canvey to carry the insulation on a suspended framework atspringing level.

Since the temperature gradient is downwards, the warmest gas rises to the highest part of the enclosed space and does not circulate. Therefore a layer of still gas provides excellent insulation. This may be either boiled-off natural gas or specially supplied nitrogen. At Port Arzew the warm free atmosphere above the gas-tight membrane provides effective additional 'insulation' for the supporting structure which is in mild steel.

Pumps, Pipework, etc.

Provision must be made for carrying the pumps and pipeworth and for passing these through the roof. The weight may be taken on the main roof framework(as at Port Arzew) on a separate structure cantilevered out from outside the tank (as at Canvey) or on a separate bridge spanning across the tank (as proposed for Stuttgart).

All pipework must have some type of provision (such as bellows or long swinging suspenders) to allow for temperature movement both of the ground, of the structure and of the pipes themselves.

In cold climates build-up of ice on the roof is inevitable. The newly boiled-off gas in contact with the underside of the roof structure must be very many degrees below freezing point (at Canvey the calculated figure is -139 deg.C.) If the outside air is damp and at 0 deg.C., however effective the insulation may be, some heat must penetrate and ice must form on the outside of the roof. The calculated figure, under adverse weather conditions, at Canvey is a build-up of 0.0134 inches of ice per hour or half a ton of ice per hour on each roof.

No trouble from ice has so far developed at Arzew.

An observed thickness of 6½ inches of ice on a 30-inch pipe adds a weight of 330 lbs. per ft. of pipe.

Roof at Port Arzew

Roof Supports

The inground reservoir at Port Arzew is 122'-0" diameter (Fig.1). It was considered safe to place the roof supports 4'-0" back from the face of the hole-a decision which gave rise to some temporary anxiety later on.(Fig.2). The roof supports therefore lie on a circle 130'-0" diameter. The ground consists of a deep stratum of stiff marl overlain by a shallow layer of weakly cemented sandstone. Above the sandstone is a few feet of sand filling. To retain this filling before the ground was completely frozen a reinforced concrete retaining wall 7'-6" high was built round the site. This wall is supported on 60 concrete foundation blocks taken down a maximum depth of 6'-0" to reasonably hard ground (Fig.2). The bottom of the wall is not attached to the foundation blocks but is free to slide over them. Rising from the top of the wall are 60 reinforced concrete columns 4'-0" high. These columns support the structural steel ring beam from which the main roof structure springs.

The roof and roof supports are designed to resist eathquake effect.

A study of ground conditions suggested that upward movement caused by freezing of the ground water would be mostly uniform with possibly a slight but uniform variation in level from one side of the site to the other. This would prove no major problem.

Fig.2 shows a section through the steel ring beam and the provision that has been made for adjusting its level. During the first two years in service no adjustment was made.

Design Data

The roof is designed to carry a positive gas pressure equal to 15 in. of water (78 lbs. per sq. ft.) and a vacuum equal to 2 in. of water (10.4 lbs per sq. ft.).

The roof structure also carries the pumps and pipework. The actual load these impose on the roof is larger than the preliminary estimate and this has caused some downward deflection of the main supporting structure.

Basic Design

The roof (Fig.1.) consists of two main parts; a flat gas-tight roof in aluminium alloy and a mild steel openwork lattice dome from which the flat roof is suspended. The flat roof is loaded with concrete to counteract the maximum positive gas pressure. The dome therefore carries its maximum load when the gas pressure is negative for it then has to carry a vacuum of 10.4 lbs.per sq. ft., concrete Kentledge weighing 83 lbs. per sq. ft., the self-weight of the flat roof (about 6lbs. per sq. ft.) and its own self-weight (about 9 lbs. per sq.ft.) making a grand total of about 110 lbs. per sq. ft.

Round the edge of the flat roof is a vertical skirt in aluminium alloy the lower end of which tucks into a sealing trench in the frozen ground (Fig.2).

Mild Steel Dome

The spherical reticulated dome is shown in Figs. 1 and 3. It has 12 main ribs running north an south and 12 main ribs running east and west and is on the Hamman

(Hamilton-Manning) principle. The ribs are spaced at 10 ft. centres on the two main centre-lines and are all placed on great circles. The plan diameter is 127'0" measured on the pitch circle and the radius of the dome is 150'0" giving a rise of 14' 1¼" or one-ninth of the pitch circle diameter. The main members are made of two 8"x3½" mild steel channels welded edge to edge to give an 8"x7" box section.

The suspenders that carry the flat roof are 1½" diameter aluminium alloy hung from the intersection points of the main ribs by universal joints. These suspenders are provided with stainless steel trunbuckles to adjust the level of the flat roof.

All ribs have adjusting screws at both ends and the ribs pass through one another at their intersections by means of a special tenon joint.

Flat Aluminium Roof

Since the suspenders that support this flat roof are placed at the intersection points of the main ribs of the dome (Figs.1 and 3) the flat roof is cut into panels that are roughly square and approximately 10'0" x 10'0". Only the panel in the centre of the roof is exactly square the other panels growing slightly smaller and slightly more oblique as they get nearer the edge (see the plan in Fig.3).

The framing consists of 7" x 3" aluminium alloy main joists spanning about 10'0" from suspender to suspender carrying 5" x 2½" aluminium alloy secondary beams spaced about 5'0" centres and spanning about 10'0". The sheeting is 14 g. (0.080") aluminium alloy in panels about 5'0"x10'0" welded to the upper flanges of the joist framing.

A ceiling of aluminium expanded metal is hung below the flat roof and carries the insulation.

Half the concrete Kentledge that holds down this flat roof against the gas pressure lies directly one the sheeting. The other half hangs on the suspenders. The panel may therefore carry a downward loading of 52 lbs. per sq. ft. or an upward loading of 36 lbs. per sq. ft.

Vertical Skirt

This skirt is in aluminium alloy and is 10'-0" high. It is scalloped in plan to accommodate thermal movement.

Roofs at Canvey

Roof Supports

The roof framing is supported by, and the roof skin is held down by, a heavy reinforced concrete ring beam (Fig.4). To comply with the main contractor's time schedule this beam was cast before the ground was frozen. One of the major problems of design was to predict and cope with the movements of this beam during construction and after the reservoir goes into service.

The surface soil is soft clay underlain by a deep bed of silt. The maximum working pressure requires a ring beam 14'-0"x6'-6" in section imposing a loading of 0.435 tons per sq.ft. on the surface which must be near the ultimate bearing capacity of this clay at zero depth. The ground was weakened by a trench to contain freezing tubes and, in one case, by a number of holes that filled with ground water, cooling brine or bentonite drilling sludge. Knowing that some uneven ground movement was inevitable the ring beam was cast in 12 separate sections each about 40'-0" long with 4'-0" gas between sections. The reinforcement from the sections projects into these gaps. It was possible to prevent these gaps closing by casting concrete diaphragms across them and possible to restrain them from opening by welding the projecting reinforcement together and by putting additional steel across the gaps. Each 40ft. section had sufficient steel to resist hogging or sagging due to possible irregular upward frost heave.

A record was kept of the levels of the 12 sections of the beam and when it appeared that relative movement had virtually ceased the gaps were concreted to form a continuous ring.

The roof construction at Canvey is sufficiently adjustable and flexible to accommodate all the general and relative movements so far encountered. Had the roof consisted of a reinforced concrete shell dome it would undoubtedly have split.

Design Data

The roofs are designed to carry a gas pressure of 38 inches of water plus 25 per cent test overload or a total of 248 lbs.per sq.ft. or a vacuum of 2 inches of water plus 25 per cent. Allowing for self-weight and possible superload the roof must carry an upward load of 240 lbs. per sq.ft. or a downward load of 36 lbs. per sq. ft.

Basic Design

The roof consists of two separate structures - a spherical skin to carry the upward pressure and a reticulated spherical dome skeleton to carry the downward loading.

Roof Skin

The gas-tight membrane that resists positive gas pressure is a spherical skin of aluminium alloy sheeting $\frac{3}{16}$" thick lying on top of the roof domed skeleton but not attached thereto. The sheets are supplied in widths of 5'0" and the shaping and site welding of these plates was one of the major problems. The edge of the skin is held down as shown in Fig.4.

Domed Roof Structure

The roof skeletons are standard type Hamman aluminium alloy domes 144'2" plan diameter on the pitch circle. They have a spherical radius of 170'0" giving a rise of 16'0½". The ribs have a basic spacing of 10'0" on the main centre-lines. The plan is therefore very similar to the plan in Fig.3 except that there are 14 ribs in each direction at Canvey instead of 12 as at Port Arzew. The main members are 5"x6" hollow rectangular extrusions with walls 0.25" thick.

These roof skeletons are assembled on an adjoining site, lifted bodily, transported and lowered into position.

The Authors have a large number of lantern slides of the work they have designed and contructed at Port Arzew and Canvey and a selection from these will be shown as time permits.

Acknowledgements

Details of the work at Arzew and Canvey are included here by courtesy of the Gas Council, the North Thames Gas Board, Conch Methane Services Limited and Sir Robert McAlpine and Sons Limited.

Reference

Details of the Hamman dome are given in "The Structural Engineer" December 1953.

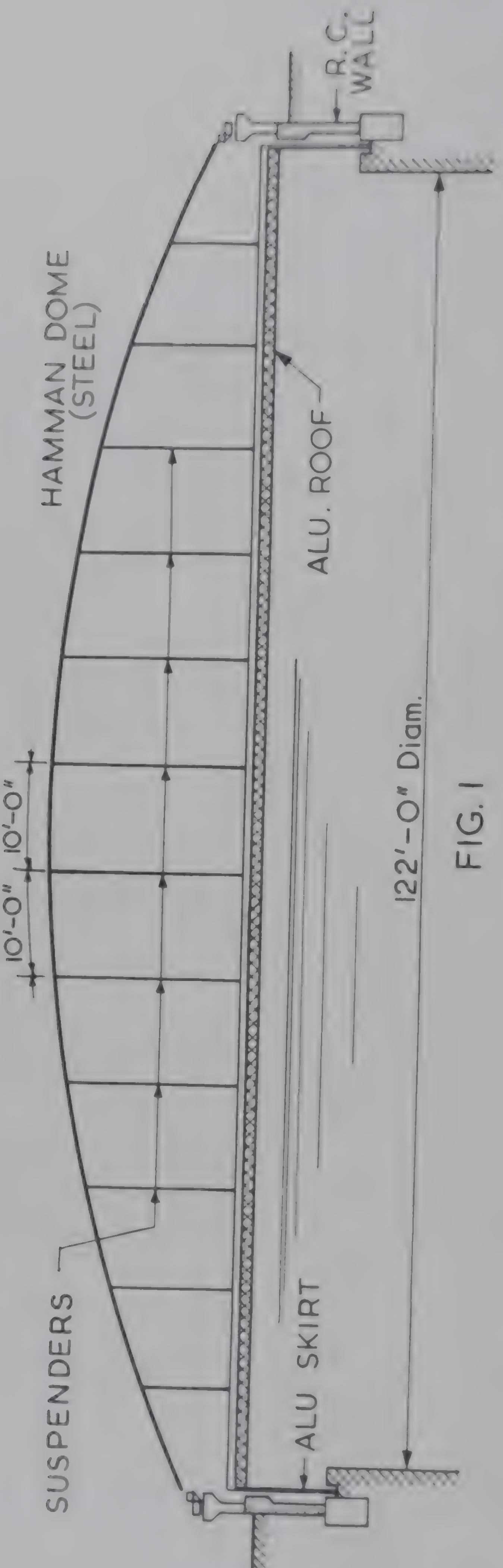

R.C. WALL
HAMMAN DOME (STEEL)
ALU. ROOF
10'-0"
10'-0"
SUSPENDERS
ALU SKIRT
122'-0" Diam.

FIG. 1

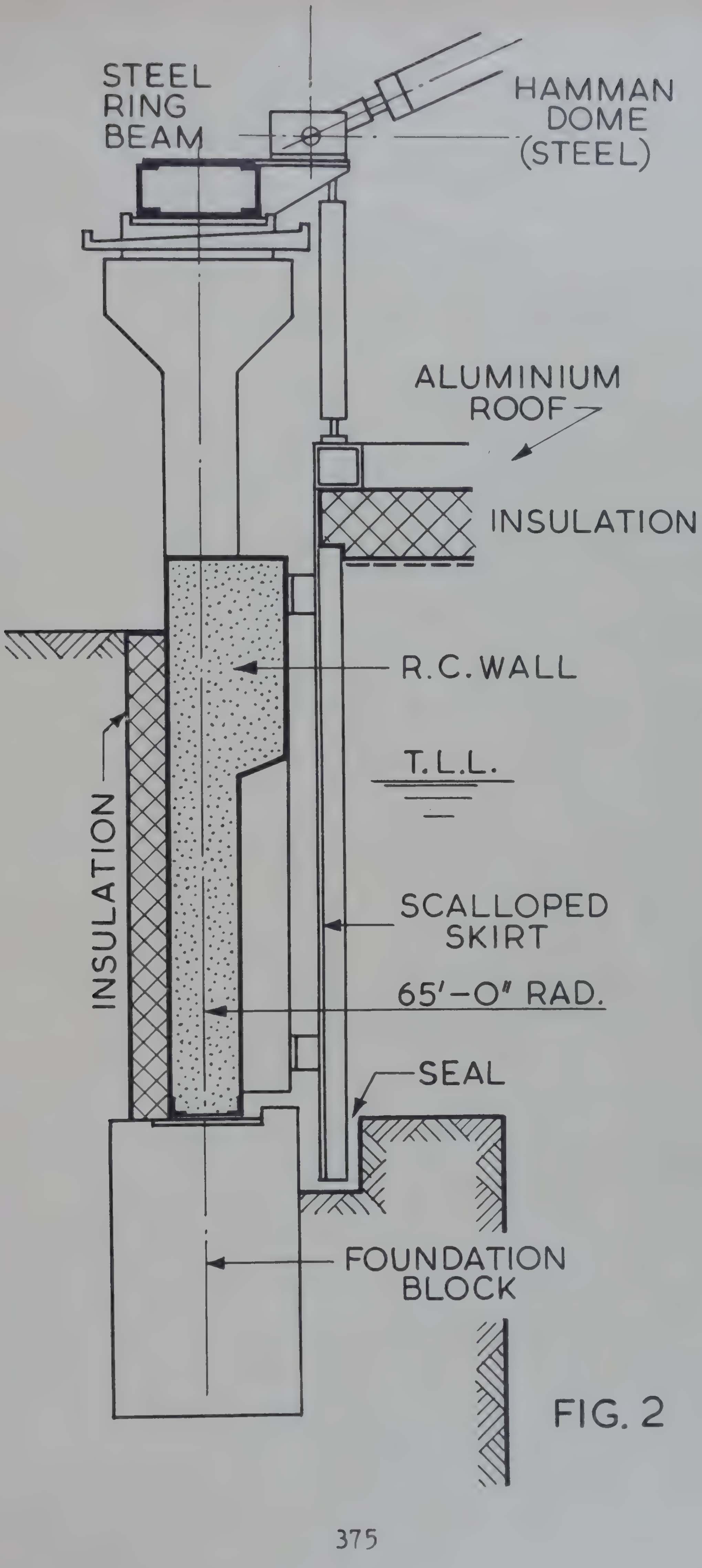
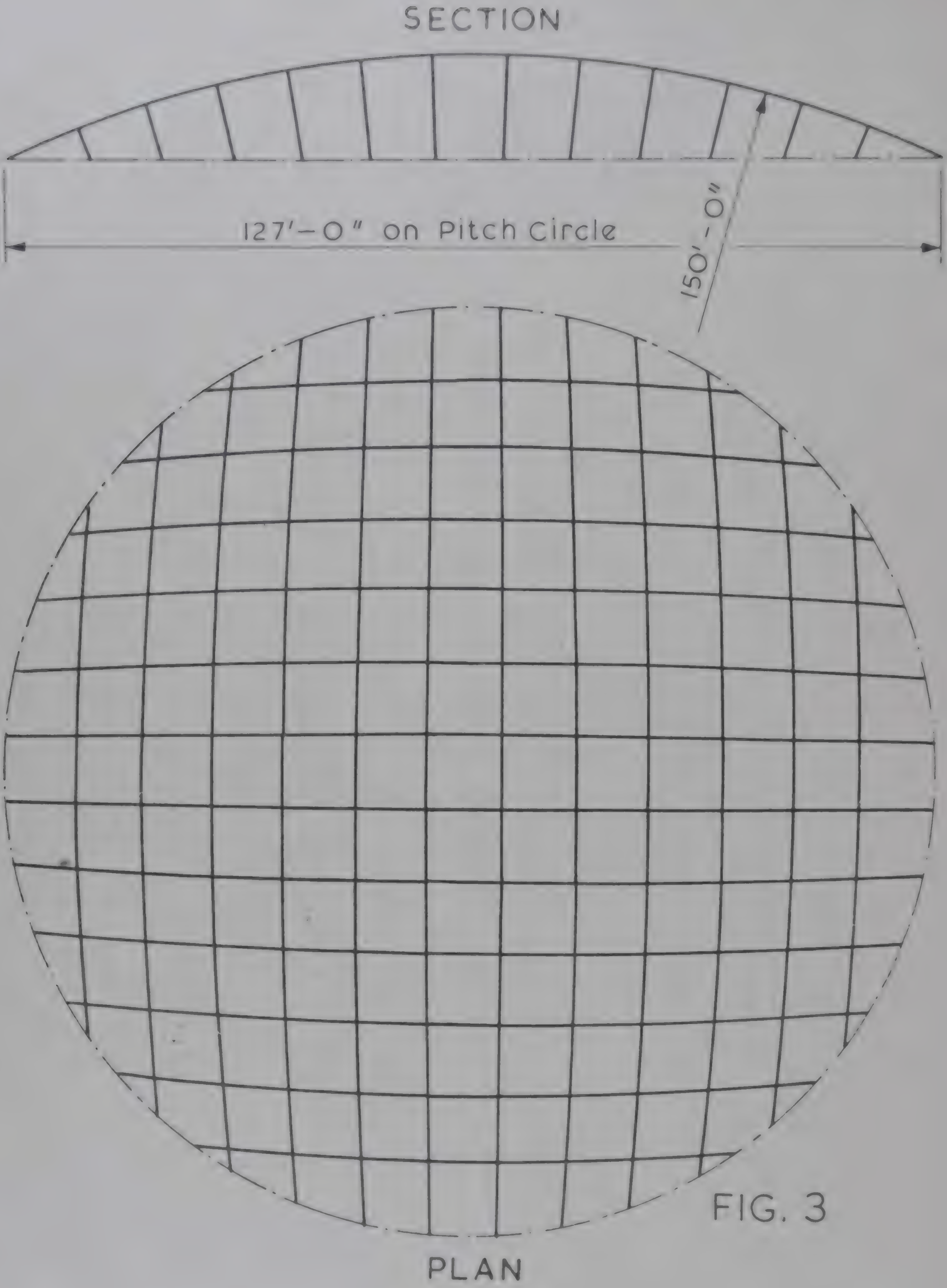

STEEL RING BEAM
HAMMAN DOME (STEEL)
ALUMINIUM ROOF
INSULATION
R.C. WALL
T.L.L.
INSULATION
SCALLOPED SKIRT
65′–0″ RAD.
SEAL
FOUNDATION BLOCK

FIG. 2


SECTION

PLAN

127'–0" on Pitch Circle
150'–0"

FIG. 3

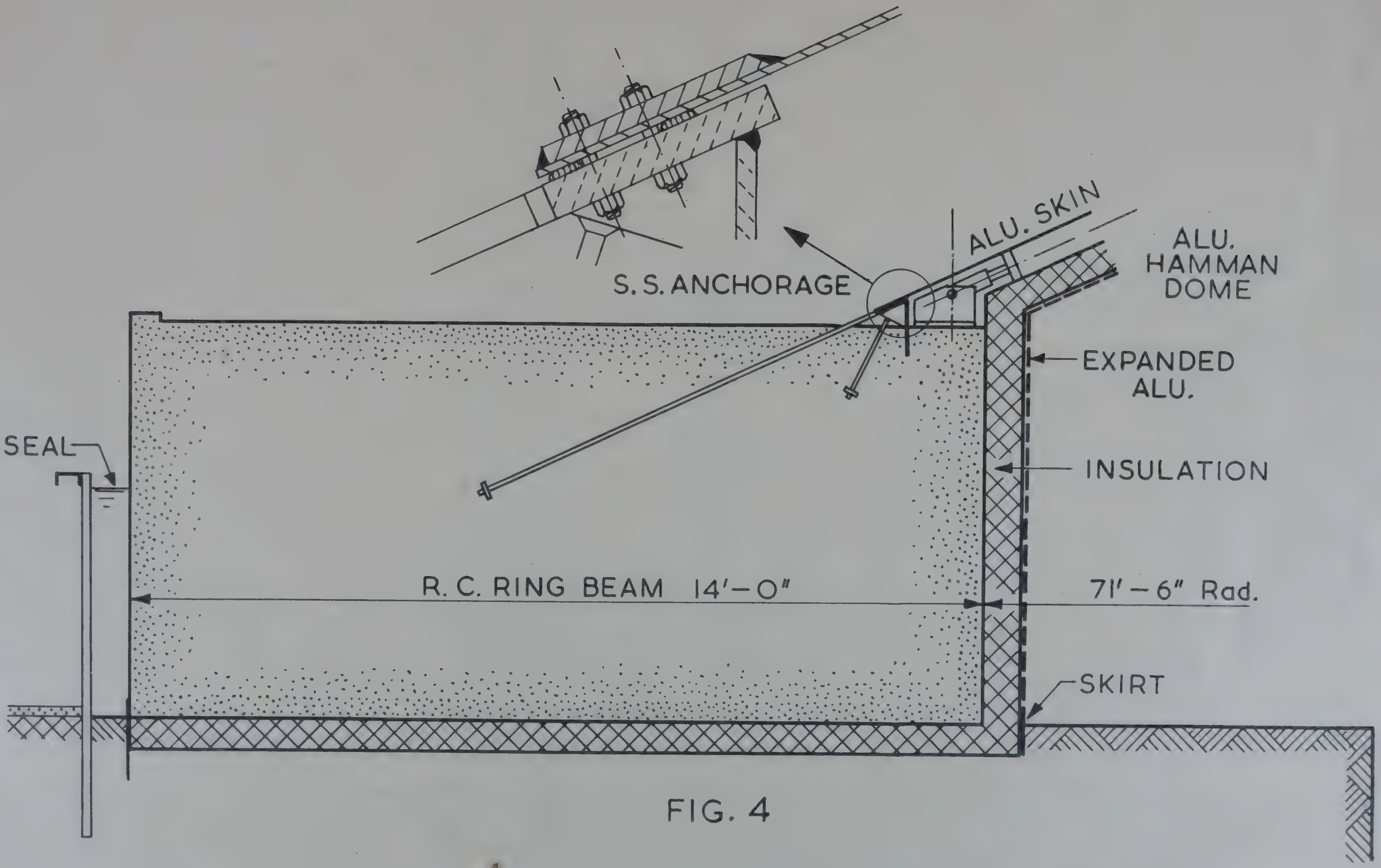

S. S. ANCHORAGE
ALU. SKIN
ALU.
HAMMAN
DOME
EXPANDED
ALU.
INSULATION
SEAL
R. C. RING BEAM 14′–0″
71′ – 6″ Rad.
SKIRT

FIG. 4

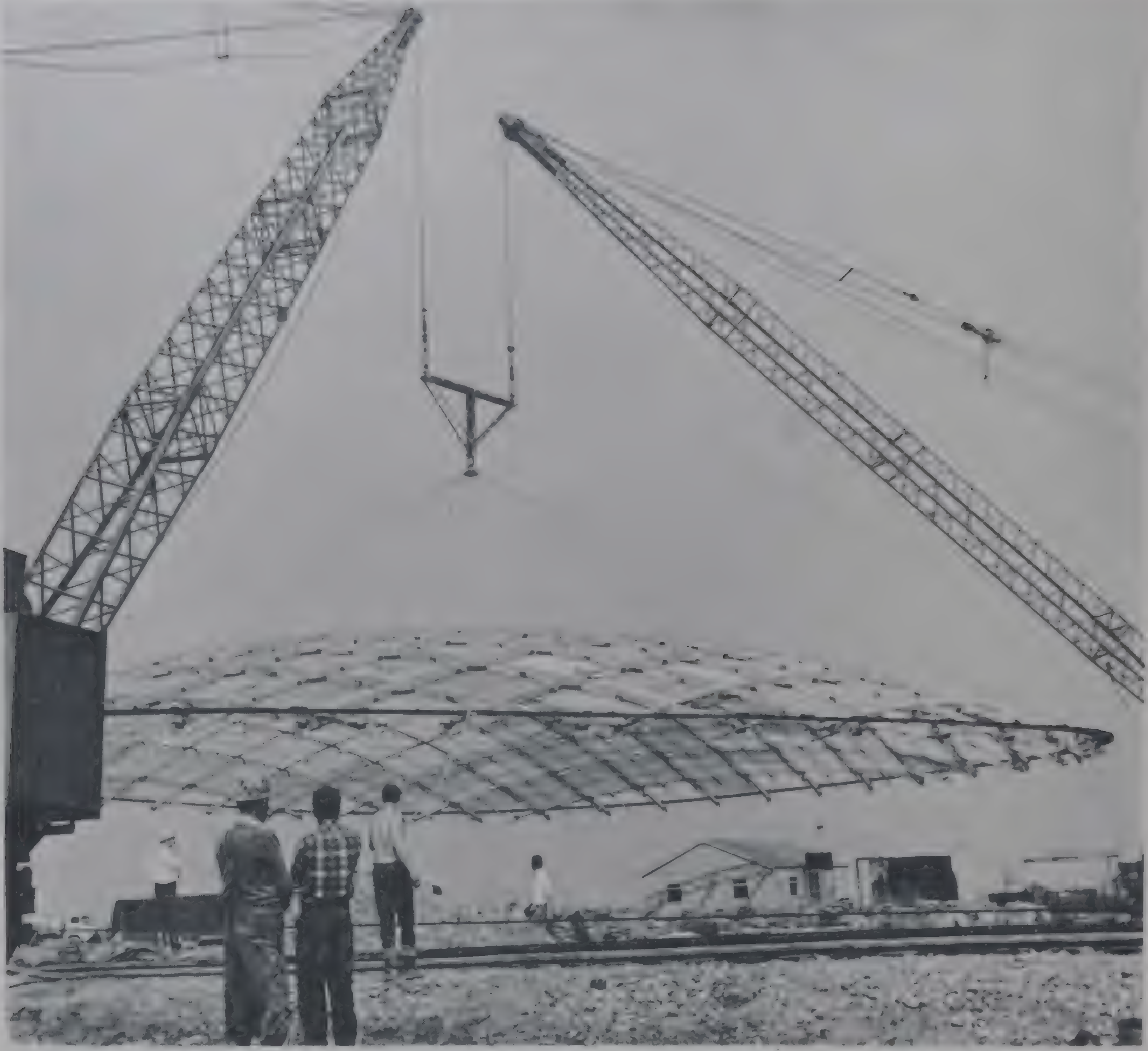


Fig.5

USE OF SUBMERGED ELECTRIC MOTOR DRIVEN PUMPS FOR LIQUEFIED GASES

By E. Hylton* and R. G. Jackson +

It is particularly appropriate to present at this conference, which deals with both LNG and low temperature and electrical power, a paper on the use of submerged electrical motor driven pumps in the transport and storage of liquefied gases.

When in 1958 the Methane Pioneer was fitted with Deepwell pumps, high speed centrifugal submerged electric motor driven pumps were already in service in aircraft fuel tanks. Although the deepwell pumps were generally satisfactory, difficulties were encountered which were believed to be due to the length of the drive shaft. The elimination of the long shaft, its support bearings and the shaft seal removes at one stroke a number of sources of potential pump failure. Additionally, the submerged electric motor principle enables the use of higher pump speeds than can be tolerated by long shaft drives.

Following careful consideration of the advantages of the two types of pump, Conch decided in 1959 to begin a series of tests to prove the suitability of the submerged electric motor driven pump for LNG service, and to provide sufficient information to the classification societies and regulatory bodies to obtain their approval for this type of pump for liquefied gas service.

Conch chose a pump manufactured by The J. C. Carter Company for this purpose. This paper describes the various steps each company has made over the last eight years in developing the pump, obtaining authority for its use and producing designs to meet the requirements found in service.

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* J.C.Carter Co., Costa Mesa, California, U.S.A.
+ Conch Methane Services Limited, London

THE BEGINNING

The pump chosen for the first tests was a standard aluminum case single stage centrifugal pump type 6366, with a rated capacity of 85 g.p.m. at 185 ft. head at 3600 r.p.m. A four-bladed inducer was fastened to the main shaft 1.5 inches above the inlet plane. The pump was fitted with ball-bearings.

The driving electric motor was a modified four-pole squirrel cage open-type non-explosion-proof induction motor with a rated capacity of 1 horsepower at 1800 r.p.m. but, by using a varidyne power supply, the frequency was varied to vary the speed so that 2 horsepower could be obtained at 3600 r.p.m. The motor was fitted with ball-bearings.

This pump was installed in an aluminum test tank at Lake Charles, Louisiana. This tank, 48.25 inches wide, 67.75 inches long, and 85.75 inches high, was internally insulated with 3 inches of balsa wood. It was situated about 300 feet from the main storage tank on the site from which liquid flowed by gravity into the test tank. The pump on test returned this liquid to the main tank.

The tests carried out were in two parts; firstly, to provide data needed to design a liquid gas pump to meet any flow and head requirements, and, secondly, to prove the suitability of the pump for use with these liquids.

The first tests carried out showed:

1. The electric motor operated for five hours in the gas phase without any increase in motor temperature.
2. After long submersion in liquefied natural gas, the pump maintained the same characteristics and efficiency.
3. After tests had shown that the motor could be operated both in the liquid and the gas phase, and that it could be restarted without any

loss of efficiency after many days' submersion, it was subjected to an endurance test during which it operated at full capacity of 85 g.p.m. for 22 hours per day for 30 days, accumulating a total operating time of 649 hours.

4. NPSH required was extremely low. Full rated delivery was achieved with less than 1 inch of the inlet suction bell submerged.

During this test, observations showed that the motor operated at liquid temperature and that the liquid near the pump was at the same temperature as liquid some distance away from the pump. After the test, examination of the motor windings showed that no change in insulation quality had taken place either visually or according to resistance measurements.

The pump was reassembled after this examination and was subsequently used to circulated propane refrigerant in the pilot scale ground storage test carried out the following year. During this experiment the pump operated for a total of 3266 hours and was submerged for 4625 hours.

In 1962 it was determined that a 60 horsepower, 2-pole, 60-cycle, 440-volt production size motor be built and subjected to a programme of thermal cycling tests. These tests were to encompass a total of 50 thermal cycles ranging between a maximum ambient temperature of 80°F and minimum temperature of -320°F. A typical thermal cycle involved heating the unit to 80°F by applying power across a single phase; when this was achieved the stator was then immediately submerged in a container of liquid nitrogen. After the boiling had subsided indicating the unit had stabilized at -320°F, it was removed from the LN_2 ambient and again heated to 80°F. Visual inspection and resistance measurements were taken following completion of the 1st, 5th, 10th, 20th, 35th and 50th cycles.

The results of the testing showed:

1. The wound stator assembly wire insulation revealed no evidence of deterioration.
2. The winding resistance showed the same measurement as it was prior to the tests.
3. The rotor cage assembly was subjected to a growler test and showed that it was unaffected by environmental tests.

CLASSIFICATION SOCIETY AND REGULATORY BODY APPROVAL

With the satisfactory completion of these tests and with the impending requirements of specifying pumps for the first commercial LNG carriers, Conch began talks with the object of obtaining approval for the pumps while The J. C. Carter Company began the design of the required commercial pumps.

Two questions had to be answered with regard to safety. On the one hand, it had to be shown that the operating conditions in the cargo tanks could be arranged so that there was no danger of an inflammable mixture being present near to the motor, while on the other hand it had to be proven that the unlikely occurrence of an electrical breakdown could not itself produce dangerous conditions.

The possibility of getting explosive mixtures in a cargo tank containing liquefied gas is practically nil since when there is any liquid in the tank there will always be a positive pressure because the liquid is boiling. There are, however, certain conditions when the tanks might go subatmospheric, e.g. during rapid pumpout during unloading, during rapid changes in barometric pressure. We were able to produce satisfactory piping and valving arrangements which, in conjunction with a liquid nitrogen supplied inert gas system, prevented air ever getting into a tank except when reaerating before a tank inspection. Steps were also taken to ensure that the pump motors were electrically disconnected except during the time of actual discharge operations. Thus a set of conditions was written into the specification of the ships which satisfied at all points the requirement that air should be excluded from the tank at all times when an electrical discharge could accidentally occur.

It is useful to note, however, that even if all these systems failed and a vacuum was created just short of the collapsing strength of the tanks, then the relief of this vacuum back to atmospheric pressure would allow insufficient air to enter the tank for the mixture to reach the explosive limit.

A second requirement had, however, to be met; that is, it had to be shown that there was no chance of electrical breakdown in the pumps - for instance, due to mechanical damage causing a local material breakdown which could itself become the source of hazard. On this question no information was available. There was, however, a programme of research underway at Queen Mary College in the University of London in which Dr. Lewis and his colleagues were studying the electrical strength of a number of liquefied gases, both hydrocarbon and non-hydrocarbon. It was possible to make some estimates of the likely behaviour of methane from the work already carried out on heavier hydrocarbons, but of necessity the characteristics of methane had to be studied. Fortunately, the apparatus in use at Queen Mary College was adaptable for use with methane and a quantity of liquid was obtained by liquefying sewer gas through heat exchange with liquid nitrogen. These tests showed that the electrical strength of liquid methane was very high and that it could, for all practical purposes, be considered as a dielectric. Figure 1 shows the breakdown voltage of liquid methane as a function of electrode spacing and it is interesting to note that it appears to have a value of 400 volts with no gap. Figure 2 shows the relationship of electrical strength with carbon hydrogen ratio. These results in conjunction with the reports and opinions of Dr. Lewis and his colleagues were submitted for approval to the various societies. Furthermore, J. C. Carter agreed to design the motor housings in accordance with U. S. Underwriters Laboratories' requirements for Class 1, Group D explosion-proof service. As a result of these various representations, approval for the use of submerged electrically driven pumps was obtained in principle from:

Bureau Veritas, in May 1961
Lloyds Register of Shipping, in November 1961
The British Ministry of Transport, in June 1962
The American Bureau of Shipping, in September 1962

PUMP DESIGN

While Conch was obtaining classification approval, The J. C. Carter Company were proceeding with the design of suitable pumps. These were to satisfy two requirements. Firstly, the pumps for shipboard applications, the particular situation which had first triggered off the studies; secondly, the design of pumps to discharge the product from the land based tanks in which it would have to be stored. The first sea-going pump was fitted to the experimental LNG Carrier "Beauvais". This was a pump of capacity 300 g.p.m. and had a delivery head of 328 feet. The first commercial pumps, however, were those fitted to the Methane Princesss and Methane Progress which had a capacity of 900 g.p.m. and a delivery head of 270 feet. A sectional drawing of this pump is shown in Figure 3. Of particular significance is the fact that a small portion of the pumped LNG is deliberately pumped through the motor bearings and stator coils providing a medium for cooling. The amount of coolant is controlled by the clearances through the impeller wear ring and flame proof passages of the motor. No rotating seals or gaskets are required since all external leakage is returned directly to the cargo tank.

At the same time, a design was produced for a range of re-gasification pumps for the North Thames Gas Board's Terminal at Canvey Island. In this instance the J. C. Carter Company determined many of the technical developments with respect to cryogenic motors and bearings on the shipboard pumps could be applied to this application. There was, however, a vast difference in the hydraulic characteristics of this group of pumps since the Canvey Terminal must maintain grid pressure of 666 to 1000 p.s.i. Each pump was designed with six stages to produce a capacity of 50 tons per hour and discharge head of 1800 feet. Installed in series/parallel with other pumps, the goal of 100 tons per hour at 666 p.s.i. was achieved. Recently additional Carter pumps were installed to provide the ultimate 1000 p.s.i. grid pressure.

The use of submerged pumps at the Canvey Terminal brought with it no special requirements. The decision, however, by Conch to commercialize their ground storage equipment at Lake Charles did raise a special requirement. A ground storage unit, by its very nature, produces difficulties for the pump engineer. Firstly, it is very deep so that the discharge pump is going to need to be well over a hundred feet below the roof. Secondly, once the storage goes into service it is impossible for anyone ever to go into it again. Thus, the discharge pump must be recoverable by means operating from outside the tank. Conch had already, in its ship pump studies, considered the possibility of self-operating valves which would allow the pump to be removed and so once again decided that the submerged pump was likely to provide the best solution. Concurrent with but quite independent of the Conch studies, J. C. Carter had also developed a removable type of deep hole cryogenic submersible pump. Accordingly, Conch began discussions with Carter who, because of the good cooperation existing, were the obvious choice to provide the specialist expertise. A design was produced as shown in Figure 4. There the pump unit is supported by a special bottom section fitted to the end of the liquid discharge line. This section not only contains a support cone for the pump, but also a spring loaded foot valve which is opened or shut by the lowering or raising of the pump unit. The pump unit is designed to be lowered or raised by the conduit carrying the electrical connections to the pump.

The first of this design of pump was installed in the ground storage unit of the liquefaction plant at Arzew, Algeria, where two pumps delivering 6000 g.p.m. each have been in operation since 1965. Patents were applied for and received by each organization covering these developments.

OPERATING EXPERIENCE

The Methane Princess and Methane Progress have been in LNG service since the Autumn of 1964. During that time the pumps have operated satisfactorily delivering slightly more than their design capacity. At each annual overhaul the pump units have been examined and the bearings of the pump and motor have been removed and, if showing any signs of wear, replaced. Although by the end of the fourth year of operation all the bearings had been replaced, this was often more due to caution than to necessity. In this period the pumps have operated more than 1000 hours each, and have discharged between them more than 2.5 million tons of LNG.

The pumps at Canvey have had the job of delivering out of storage into the national gas grid this same quantity of LNG. In this period the Canvey booster pumps have operated in excess of 14,000 hours. As would be expected in a system of this type, problems of a minor nature have developed; however, the operation is now largely of a routine nature.

The two large pump units at Arzew installed in such a special way because of their inaccessibility have, as might be expected, provided the most trouble. Both pumps have had to be removed twice from the storage because of electrical faults of the type that some classification societies concerned themselves in the early days. The first of these was caused by contaminants which had found their way into the conduit while they were stored somewhat unsatisfactorily prior to installation at Arzew. These contaminants produced a short circuit in the conduit tube which caused the tube to break and lead to the first "pump fishing operation". Although the conduit was broken more than twenty feet from the top, a makeshift tool was able to fish it out far enough for the normal removal sequence to be operated. Examination of the fracture showed that severe arcing had been taking place before the fracture but that this had led to nothing more dangerous than the fracture of the tube.

The next failure was, however, from a technical point of view, much more spectacular. After 243 hours of operation, while pumping liquid to one of the ships, a fault developed in the motor. This happened in such a way that the contact breaker on the faulty phase did not come out and the contacts fused together, thus holding the breaker permanently in place. The motor continued to run on two phases until it became so hot that the aluminum rotor bars began to melt. The motor, of course, stopped itself. Once again nothing happened except to the motor. These accidents vindicated very well the opinions of the experts who had supported us in our approaches to get the pumps accepted.

Twice more the pumps have had to be removed due to fatigue failure of the weld of the bottom flange of the conduit. In one of these failures the pump had to be removed by a fishing operation from the bottom of the discharge column, a feat which was well achieved by the engineering personnel of Camel (see Figure 5 showing the pump arriving at the top of the column supported by its fishing line). Apart from these faults the pumps have performed above their design capacity throughout. Examination of the whole of the units during the electrical repairs showed that the rest of the pump was in excellent condition. It is worthy to note that the pump aluminum castings appear to be in "as new" condition with no indication of corrosion of any kind. We were also able to observe a like condition on the shipboard pumps after four years of submergence and operation.

FURTHER DEVELOPMENT

In the four years since the Arzew to Canvey scheme went into operation, many more pumps have been made and sold for LNG service. One innovation during this time has been to adapt the ground storage type pump to shipboard operation. Since the initial installation of J. C. Carter cargo pumps on the Methane Princess and Progress, there are now twenty-three semi-refrigerated and fully refrigerated liquefied gas carriers in operation with Carter fully submerged cargo pumps. A total in excess of 350 units have been delivered to date.

The authors of this paper feel this is an excellent example of cooperation between user and manufacturer where a definite need had been established and development response was forthcoming through the joint technical resources of both organisations.

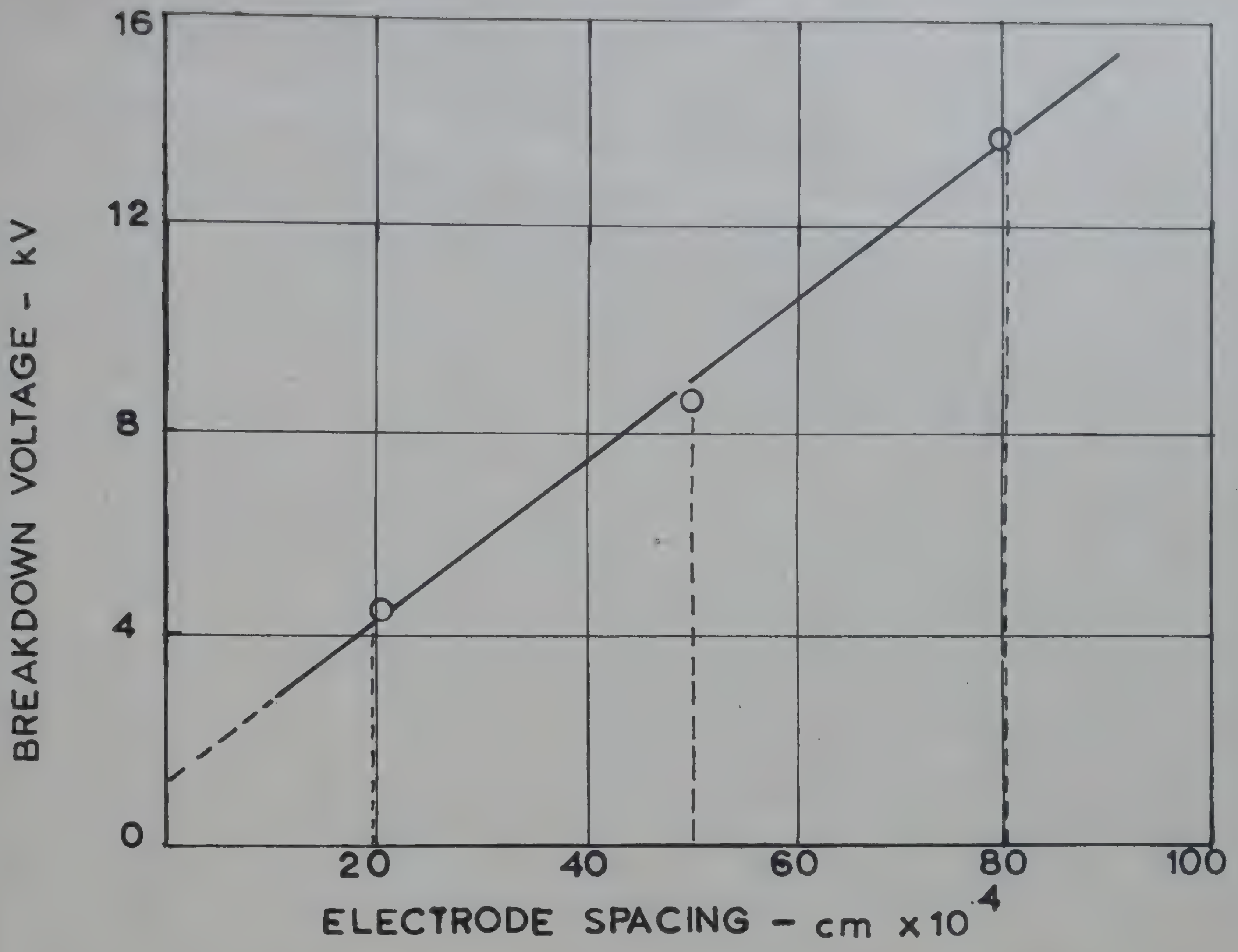


Fig. 1. Breakdown voltage against electrode spacing.
Stainless steel electrodes, zero O.P.

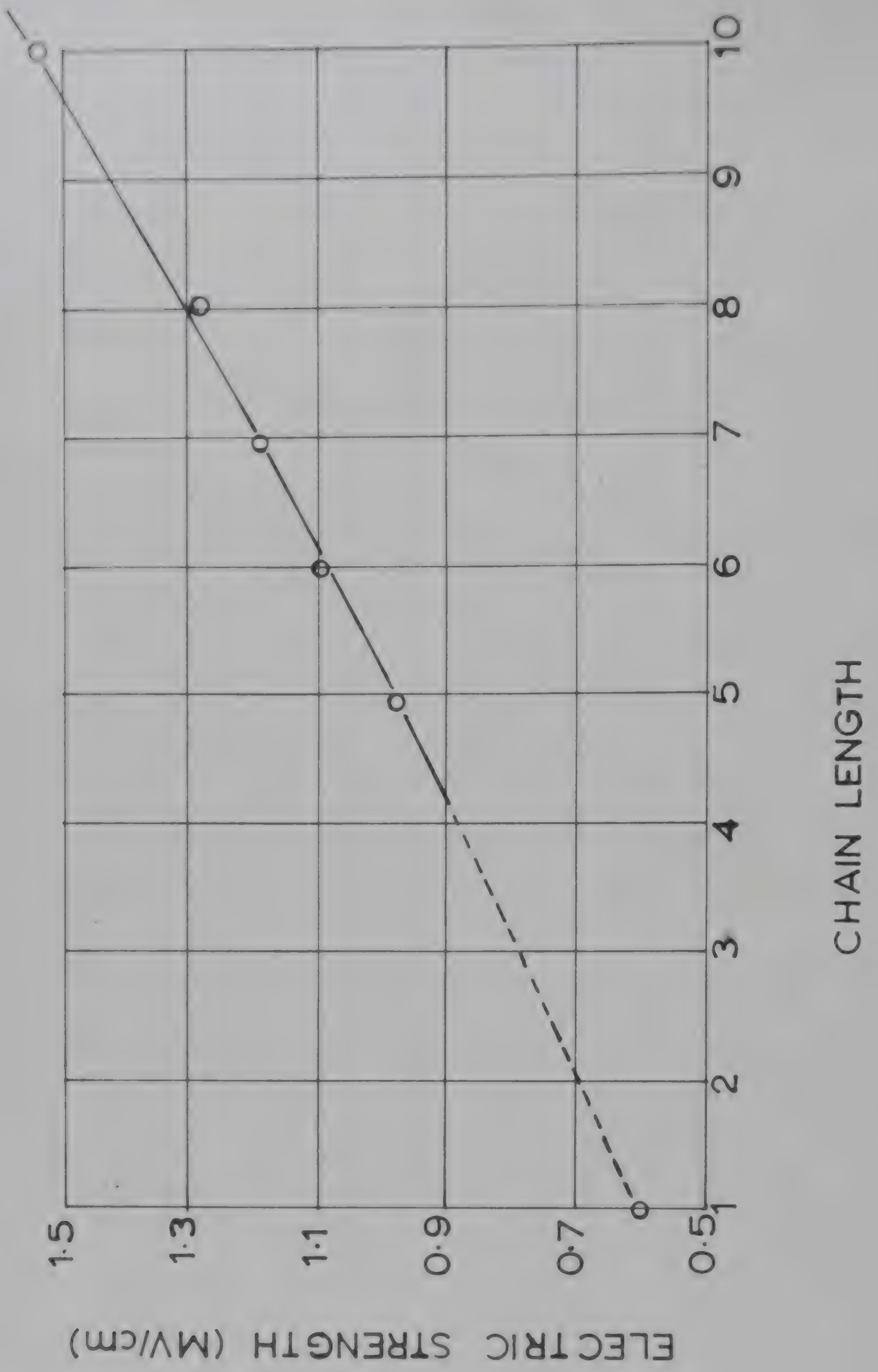
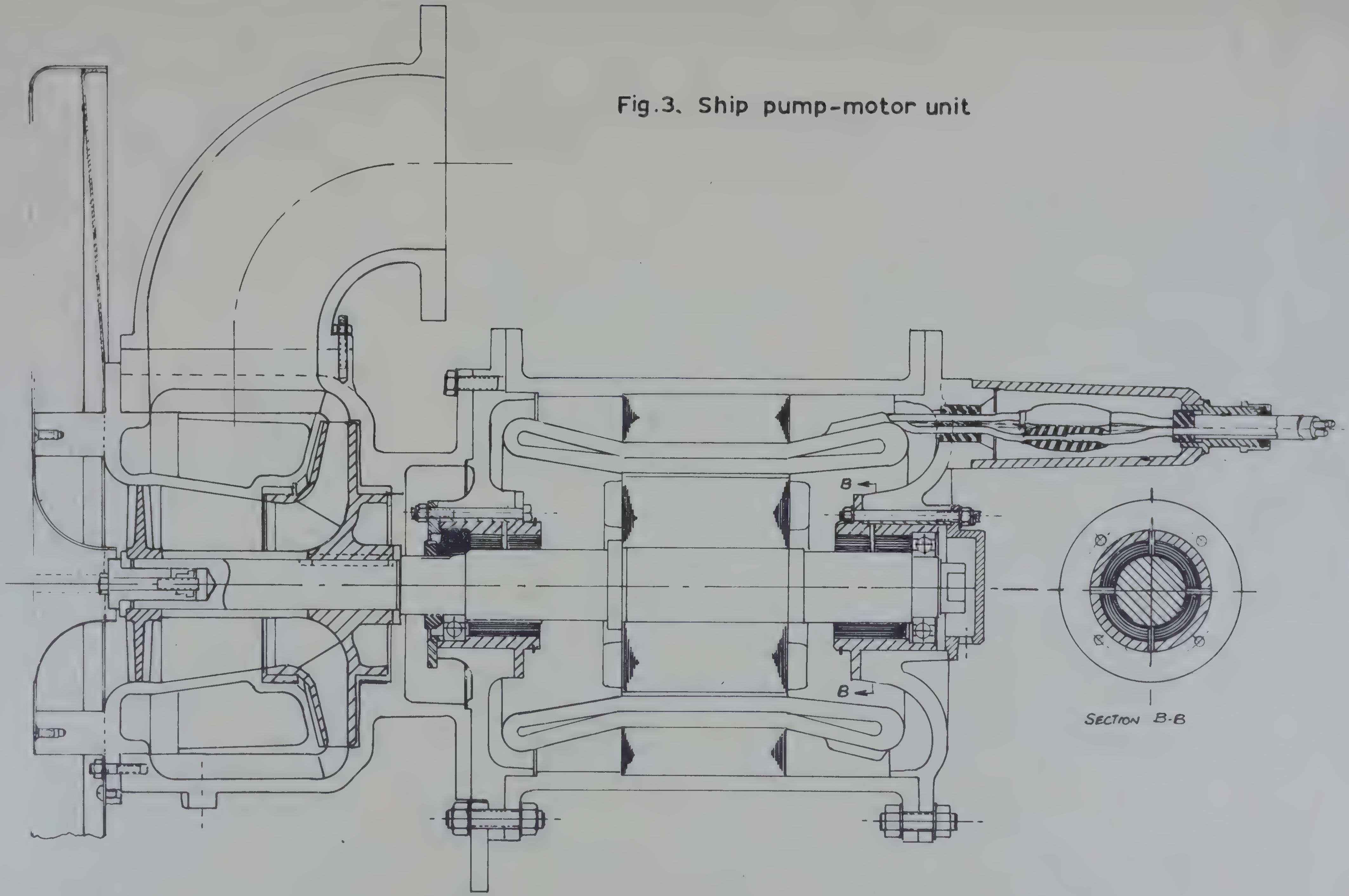


Fig. 2. Variation of electric strength with hydrocarbon chain length

Fig.3. Ship pump-motor unit


B
B
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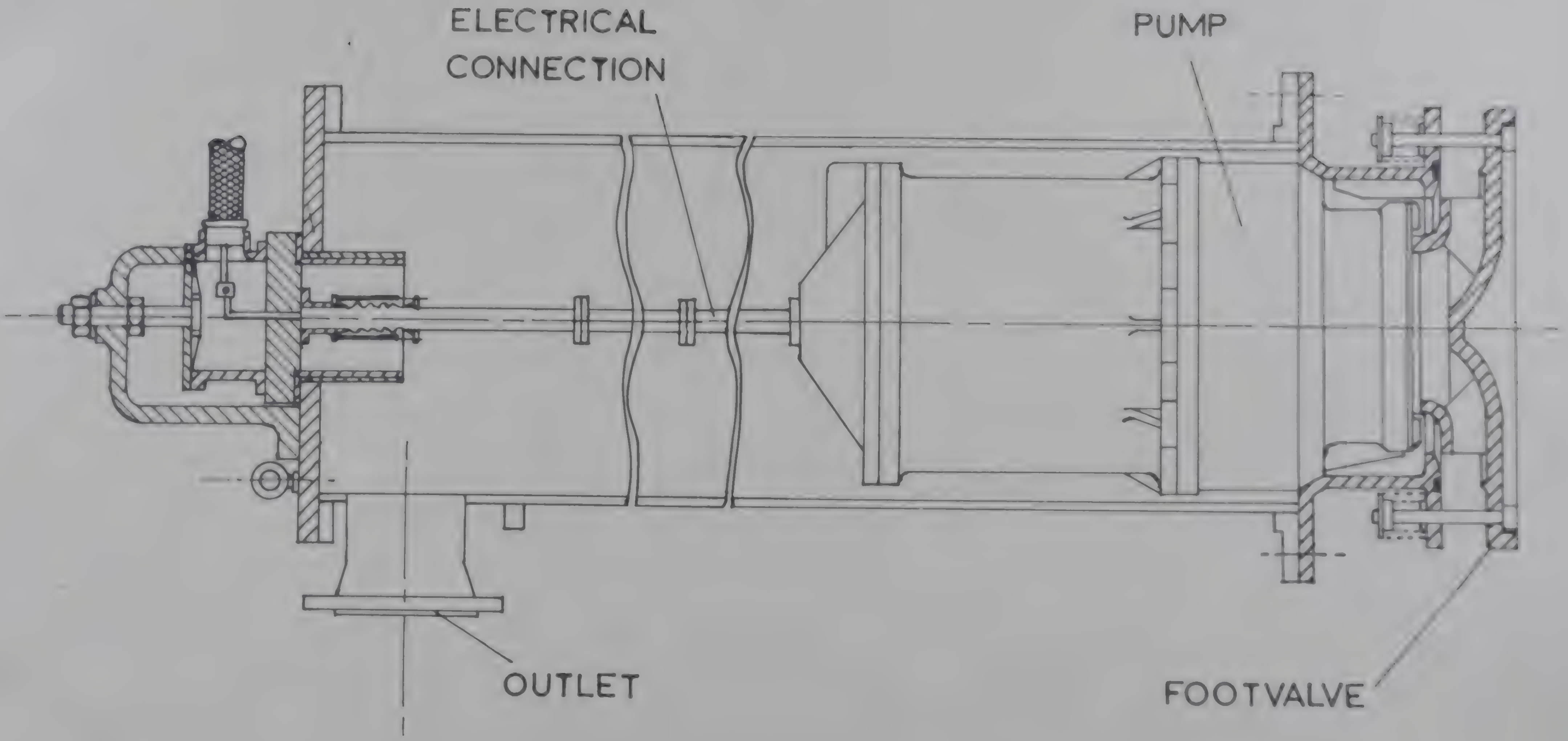

ELECTRICAL
CONNECTION
PUMP
OUTLET
FOOTVALVE

Fig. 4. Removable pump-motor unit.

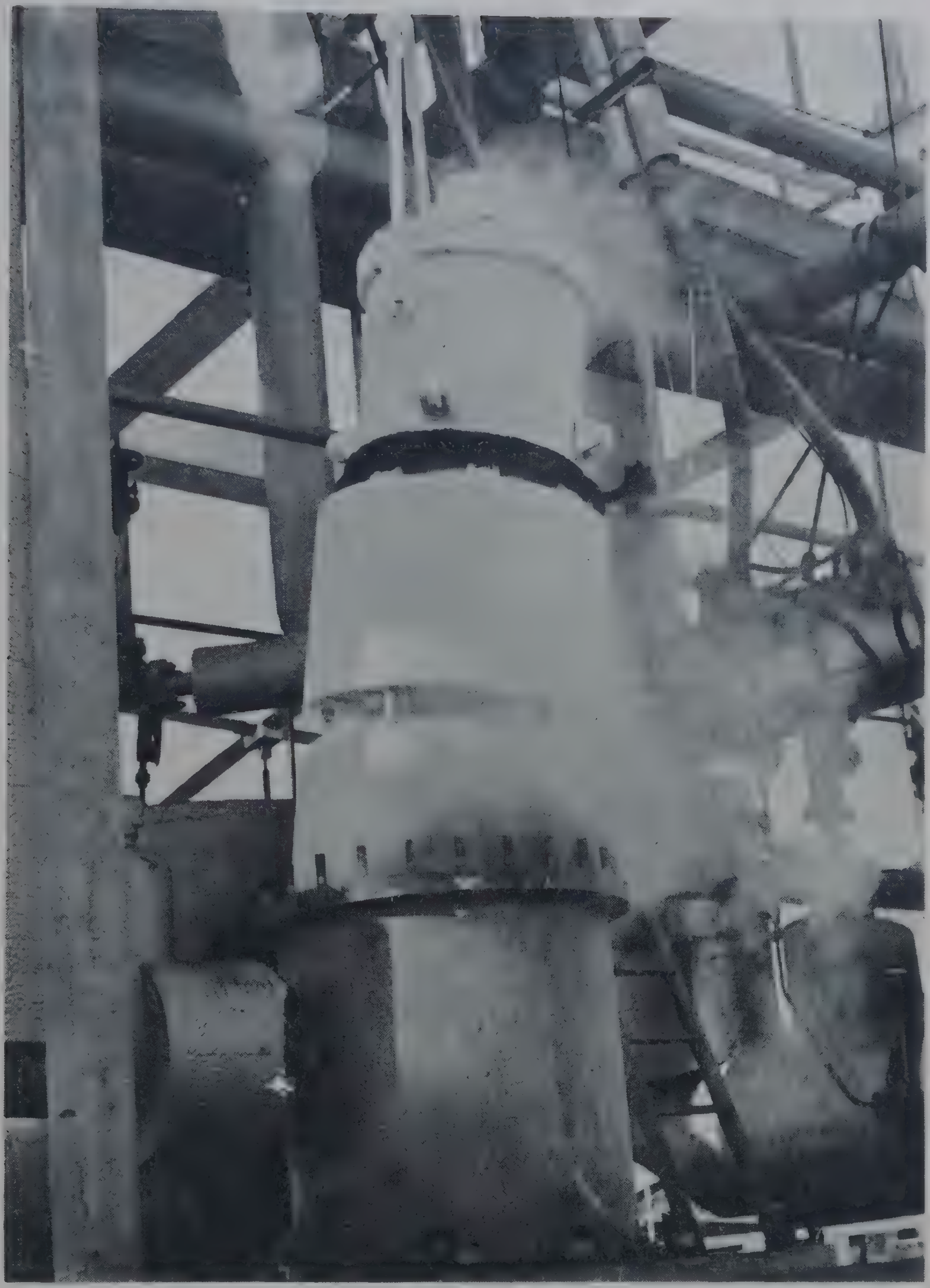


Fig.5. Pump "fishing" operation

Problems in connection with the foundation of tanks containing a lower temperature media

E. Zellerer, LINDE A.G. München

1. Introduction

In connection with the foundations of tanks containing a media whose temperature is equal to the surrounding temperature, the following are the determining criteria:

a) the permissible bearing capacity (to guard against foundation failure) and the

b) settlements expected.

For the tanks holding a media of lower temperature the influence of the cold upon the soil (fig. 1) has to be taken into account as well. In that connection a basic distinction has to be made between

a) soils which are not susceptible to frost action and

b) soils which are susceptible to frost action only under certain hydrological conditions and

c) soils which are always susceptible to frost action.

While dealing with the frost susceptibility the question of the depth of the frost penetration arises. The frost depth is dependent upon:

1) the temperature of the medium stored
2) the temperature of the soil
3) the co-efficient of thermal conductivity of the soil
4) the co-efficient of the heat transmission of the insulation and
5) the diameter of the tank.

In the paper presented here, the various influences are studied diagrammatically. For purposes of illustration the extreme measured values for the co-efficient of thermal conductivity of the soil have been used. For insulation the co-efficient of heat transmission generally used in practice were adopted. Further the influence of the bottom heating is examined whose extent is also dependent upon the influences already mentioned.

The paper is the outcome of a current development and research programme of LINDE A.G., Munich.

2. Frost criteria

Coarse grained soils such as gravels and sands are not susceptible to frost action. In such soils an increase of about 9 % takes place while freezing. The extra volume finds it place in the voids between the grains. Also because of the lack of capillary systems in such soils they are incapable of drawing in additional water from adjoining sources.

Soils of this type therefore are not susceptible to the influence of the cold and inspite of the formation of ice lenses there is, as a rule, no heaving of the structures.

By contrast, the so called cohesive soils like loams, silts, clays as well as gravels and sands interspersed with loam, in relativly dense state, are susceptible to frost action. Above all, these soils show a ramified capillary system. During the process of freezing, the water is drawn towards the center of freezing from the surrounding areas, as well as from the lower subsoil which leads to the formation of ice lenses. The resulting heaving pressure can be of such an intensity as to cause lifting of even very heavy structures and endangering their stabiltiy.

For judging the frost susceptibility of a soil in practice the criterium of A. Casagrande is applied. According to this the share of the fine fractions with a grain size $< 0{,}02$ mm determines the susceptibility to frost. The grain analysis takes into consideration only the share of grains $< 2{,}0$ mm.

Depending upon the percentage of fine grain fraction the well graded soil, i.e. soils which contain grains of all sizes are more susceptible to frost action than soils having a uniform size. Depending upon the size of the so called uniformity co-efficient the soils having 3 to 10 % share of grains $< 0{,}02$ mm are susceptible to frost.

Generally any noncohesive soil, independent of the degree of uniformity, has to be considered as susceptible to frost if it contains more than 3 % by weight of grain sizes $<$ 0,02 mm.

Whether the frost heaving will occur depends upon the water supply from the surrounding and subsoil strata.

The degree of the susceptibility to frost can only be determined in a soil mechanics laboratory.

3. Considerations of design

For the design of the foundations it has to be determined whether the penetration of cold into the subsoil is to be prevented or not. From the point of view of the Chemical Engineer the evaporation losses have to be considered while from a point of view of the Civil Engineer the frost action is of paramount importance. It is possible here that because of freezing, irregular heavings occur which max cause damage to the tanks and may endanger the stability of the structure, for example due to buckling of the tank walls.

The penetration of the frost may be prevented for instance

1) by the arrangement of a heating system under the floor of the tank

2) application of a special fundament construction along with ventilation channels.

Disadvantages of the heating system are:

a) running costs of electricity,
b) loss of economy of operation due to evaporation of the medium corresponding to the amount of heat applied
c) danger of failure of electricity or damage to the heating system for example as a consequence of unforeseen settlements.

Disadvantages of the construction with ventilation channels are:

a) Costly reinforced concrete construction
b) Loss of economy of operation due the evaporation of the medium corresponding to the warmth from the air current
c) Dependence of the functioning of the air current on the geographical position
d) on the position in regard to adjoining buildings and
e) to the main direction of the wind
f) Dependence upon the air temperature which may be at least temporatily below zero.

Even when the temperature is above 0° C, the condensation point can so lie as to result in the formation of ice in the ventilation channels, thereby leading to its mal function.

These problems seem to justify the question: Under which circumstances is it possible to omit the above mentioned measures and to permit the flow of the cold directly into the subsoil? A profitable side effect would be the insulating effect of the soil.

4. Depth of the penetration of the frost into the soil

4.1 Assumptions

As a rule the bottom of the tanks is provided with an insulating layer of varying thickness (for instance of foamglass). If one omits further measures (like heating or ventilation) freezing of the soil will occur in spite of the insulation. Because in due time the insulation layer is permeated by the 0^{o} - Isotherm and the cold then penetrates into the subsoil without any hindrance.

With the customary economical layers of insulation this occurs in a relatively short time.

The depth of the frost penetration is dependent upon various factors which will be presented along with diagrams and discussed in the following paras. The idealized boundry conditions and assumptions are:

1) The soil layer is of infinite thickness and has a uniform co-efficient of thermal conductivity

2) The flow of heat, after a passage of sufficiently long interval of time is stationary.

3) The ground surface is at the same time a heat flow line, that is there is no exchange of warmth between the surface of the soil and the atmosphere.

Designations

ϑ_I = Temperature of the medium stored in C°

ϑ_{II} = Temperature of the soil, assumed for temperate zones to equal 10° constant

d = thickness of bottom insulation in m

t = depth of the 0°-Isotherm (depth of maximum frost penetration) in m

r = radius of the tank in m

λ_i = co-efficient of thermal conductivity of insulation (kcal/m h C°)

λ_B = co-efficient of thermal conductivity of the soil (kcal/m h C°)

$K = \frac{\lambda_i}{d}$ = co-efficient of heat transmission (kcal/m^2 h C°)

4.2 Tank bottom without insulation

In this case the penetration of cold into the subsoil takes place immediately. Here the depth of the frost penetration is independent of the co-efficient of thermal conductivity λ_B of the soil and is calculated by the equation:

$$t = \frac{r}{\tan \frac{\pi \cdot \vartheta_{II}}{2(\vartheta_{II} - \vartheta_I)}}$$

As can be seen from fig. 2 and 2a that the depth of the frost penetration increases lineally with the diameter of the tank and with the difference between the temperature of the stored medium and the temperature of the undisturbed soil.

Thus for a tank diameter of 20 m the frost penetration reaches to a depth of 27,5 m with ammonia but with LNG as much as 110 m.

In fig. 1 the ideal theoretical form of the ice lense is presented which corresponds to a rotation ellipsoid. This visualisation will be required for tanks placed next to each other or for adjoining structures for the accessment of mutual influences.

4.3 Tank bottom with insulation but without heating

Here the insulation of the tank bottom prevents the direkt uninhibited flow of the cold into the subsoil. Therefore the thermal conductivity of the soil in relation to the conductivity of the insulation determines the position of the 0^{o}-Isotherm in contrast to the condition described earlier.

The smallest depth results in a soil of good conductivity, for example weathered moist gravel and at the same time high resistance of the insulation and vice versa.

The depth of the frost penetration (0°-Isotherm) may be calculated here from the relation:

$$t = \frac{r}{\tan \frac{\pi \cdot \vartheta_{II}}{2(\vartheta_{II} - \vartheta_o)}}$$

with the auxiliary value = *temperature at the insulation level*

$$\vartheta_o = \frac{2 \cdot \vartheta_{II} \cdot \lambda_B + \vartheta_I \cdot K \cdot \pi \cdot r}{K \cdot \pi \cdot r + 2 \cdot \lambda_B}$$

Fig. 3, 3a, 4 and 4a depict the relationship between the frost penetration depth and the co-efficient λ_B of thermal conductivity of the soil, the co-efficient of heat transmission K, of the insulation, the temperature of the medium stored and the diameter of the tank.

Fig. 3 applies to Ammonia with $\vartheta_I = -35^\circ$ C and a tank radius $r = 10$ m ($d = 20$ m)

Curve (1) shows the frost penetration depth in a soil with poor co-efficient of thermal conductivity of $\lambda_B = 0{,}22$ kcal/m h C°
(for instance dry sands).

For insulation with a value of $K = 0{,}10$
value of $t = 24{,}0$ m

Curve (2) shows the frost depth in a soil with good co-efficient of thermal conductivity of
λ_B = 2,00 kcal/m h C°
(for example wet gravels, loams)

For insulation with a value of K = 0,10
value of t = 10,0 m

It can be seen that the influence of the type of soil is considerable.

With an insulation of K $\leqq$ 0,033 and soil having good conductivity no frost will penetrate into the subsoil. From the point of view of economy the results are very remarkable since for such a case no bottom heating would be required.

For λ_B = 2,00, K = 0,06 and ϑ_I = -35° C fig. 3a shows in addition the relationship between frost penetration depth and the diameter of the tank. The function is almost a straight line and below the intersection point at r = 7,8 m (d = 15,6 m) no ground frost occurs.

Fig. 4 applies to Methane with ϑ_I = -165° and as in previous figure the tank radius r = 10 m.

Curve (1) shows the frost penetration depth for a soil with λ_B = 0,22 kcal/m h C°
for short term storage container with
K ~ 0,05 : t = 86,5 m
for long term storage container (peak shaving)
K ~ 0,02 : t = 65 m

Curve (2) - soil with $\lambda_B = 2{,}0$ kcal/m h C^0
for short term storage container with
$K \sim 0{,}05$: t = 30,5 m
for long term storage container (peak shaving)
with $K \sim 0{,}02$: t = 12,5 m

If the insulation of the tank bottom is omitted an ultimate value for the frost penetration depth works out to be = 110 m (with d = 20 m tank diameter).

Fig. 4a shows, that for small tanks up to a diameter of d = 6,0 m with a comparatively thick insulation of K = 0,03 there is no penetration of cold into the subsoil.

4.4 Tank bottom with insultation and heating

Here it is possible to fix the postion of the 0^0-Isotherm in such a way that the frost penetration does not progress below a certain depth. Depending upon the soil conditions, the deeper the frost penetration allowed, the smaller is the heating power required.

In order to calculate the depth of frost penetration for the case of tank bottom with insulation and heating, the considerations outlined above apply as well, with the difference that instead of temperature ϑ_I of the medium as per para 4.2 and its corresponding fig. 2, the value of temperature ϑ_0 prevailing at the insulation level is to be used see fig. 5. For example for a tank of 10 m radius the depth of the frost penetration independent of the temperature of the medium works out to be as follows:

For $\vartheta_0 = -5^\circ$ C to $t = 5,8$ m
$\vartheta_0 = -10^\circ$ C to $t = 10,0$ m
$\vartheta_0 = -20^\circ$ C to $t = 17,3$ m

With decreasing temperature ϑ_0 the losses of heat for the tank bottom decrease in relation to the temperature difference $\Delta\vartheta = \vartheta_{II} - \vartheta_0$ which corresponds to a proportional decrease of the energy for heating and reliquifaction.

For tank-medium temperature $\vartheta_I = -165^\circ$ C and soil temperature $\vartheta_{II} = +10^\circ$ C.

Fig. 6 shows the percentagewise decrease of the required heating power as a relationship between the temperature of the soil and the radius of the tank. The diagrams are based on the following constants:

co-efficient of thermal conductivity of soil
$\lambda_B = 0,22$ and $2,0$ kcal/m h C°
co-efficient of heat transmission of insulation
K = 0,02 and 0,05
temperature at the insulation level
$\vartheta_0 = 0^\circ, -10^\circ$ and -20° C.

According to these figures, the savings in heating power required, increase with an increasing co-efficient of thermal conductivity of the soil and increasing depth of the frost penetration and increasing heat resistance of the insulation.

If the 100 % borderline is exceeded the heating may be omitted. This is the case for example in

curve (5) with a tank radius $r \leq 13\,m$ and $\vartheta_0 = -20^\circ$ C,
$\lambda_B = 2{,}00$ (soil of good conductivity)
$K = 0{,}02$

and also in

curve (4) with a tank radius $r \leq 8$ m, $\vartheta_0 = -10^\circ$ C
otherwise as above.

The heating energy can be calculated after the formula without consideration of the heat of the soil

$$q_{hx} = K\,(\vartheta_0 - \vartheta_I) \text{ and}$$

when taking into consideration the heat of the soil

$$q'_{hx} = K\,(\vartheta_0 - \vartheta_I) - \underbrace{\frac{2 \cdot \lambda_B\,(\vartheta_I - \vartheta_0)}{r \cdot \pi}}_{\text{heat of soil}}$$

As a result of the insulation effect of the soil, the saving therefore is:

$$\Delta q = (q_{hx} - q_{hx}')/q_{hx} \times 100\ [\%]$$

As the entirety of the curves shows, savings can be obtained by regulation of the temperature at the insulation junction.

With tanks of d = 40 m diameter and favorable soil conditions the savings may reach up to 65 %.

5. Conclusions

It has become evident, that the inclusion of the soil into the concept of the foundation may bring about considerable savings in the cost of energy or in the cost of bottom insulation. This is because the lower the temperature ϑ_0 at the insulation level is, the smaller is the evaporation loss. Hence smaller can be the expensive insulation at the bottom.

Under certain circumstances it is even possible to omit heating at the bottom altogether.

A precondition for this however is an exact knowledge of the soil conditions and their co-efficients of thermal conductivity. Above all one has to know if the formation of the ice lenses is possible, i.e. whether layers susceptible to frost action and the supply of water necessary for the growth of the ice lenses are present. In addition the position and the thickness of these layers are of interst.

It is most unfavorable if the layers susceptible to frost action are situated at the top and of great thickness since then the stationary position of the 0°-Isotherm has to be fixed in the insulation which requires corresondingly a higher cost of energy.

The question arises whether when a tank is empty, the formation of ice lenses can be reduced by applying an external load as a surcharge and further if an ultimate load exists at which the formation of ice lenses can be prevented altogether. On account of the financial importance of this question a research programme in regard to it is at present being conducted at the soil mechanics laboratory of the T.H. Munich under the direction of Professor Jelinek, at the request of the LINDE A.G. Werks, Munich. First results have shown in fact that with increasing load the rate of heaving decreases.

The ultimate pressure will occur when the surcharge is greater than the suction load created during the formation of ice lenses and thereby arrest the formation of the ice lenses. This ultimate pressure is probably identical with the heaving pressure with obstructed growth and lies according to the type of soil between 1 and 20 kp/cm^2.

(This is an accord with other known research works where for special cases values of 7 and 11 kp/cm^2 were obtained).

It is presumed that before the value of ultimate load is reached, there exists a stationary value of the rate of heaving, even in the case of higher loads. The effect of the load on the sample is presented as the ratio between values of rate of heaving of loaded samples to that of unloaded samples. As approximation the values existed between 1:2 and 1/20.

It was further observed that especially with higher loads the growth of the ice lenses set in only after delay. This response time could mean that a soil under a surcharge load at first appears to be lacking a tendency to form ice lenses, could under a prolonged exposure to frost formation start building ice lenses. In the case of storage of liquid gases such a prolonged exposure is foreseen.

Finally it remains to be ascertained if under a constant load the coarse grained soils allow building of the ice lenses more slowly than in the case of fine grained soils.

If the layers susceptible to frost action lie deeper and are overlain with a soil not susceptible to frost action then the 0^o-Isotherm can be made to lie into the corresponding depth.

For that reason a considerable reduction of the cost of energy may be possible. Under certain circumstances the heating may be entirely left out.

Otherwise the reflections outlined above regarding the influence of the surcharge upon the ice lenses formation, apply, while here the surcharge ist augmented by the non-susceptible-tofrost overlying soil. In addition due consideration should be given to the fact that through the load distributing effects of the overlying strata (if of sufficient thickness) the heavings that may occur are even and non damaging.

If in the upper zone smaller layers susceptible to frost exist, the question arises whether these may be traversed by the frost without damaging effects to the stability of the tank construction. The results of the research work at the T.H. Munich as mentioned above show that there exists a connection between the rate of change of the 0°-Isotherm and the formation of the ice lenses and that the latter may be prevented altogether under certain circumstances when the rate of change is fast enough so that there is not enough time for the ice lenses to be formed.

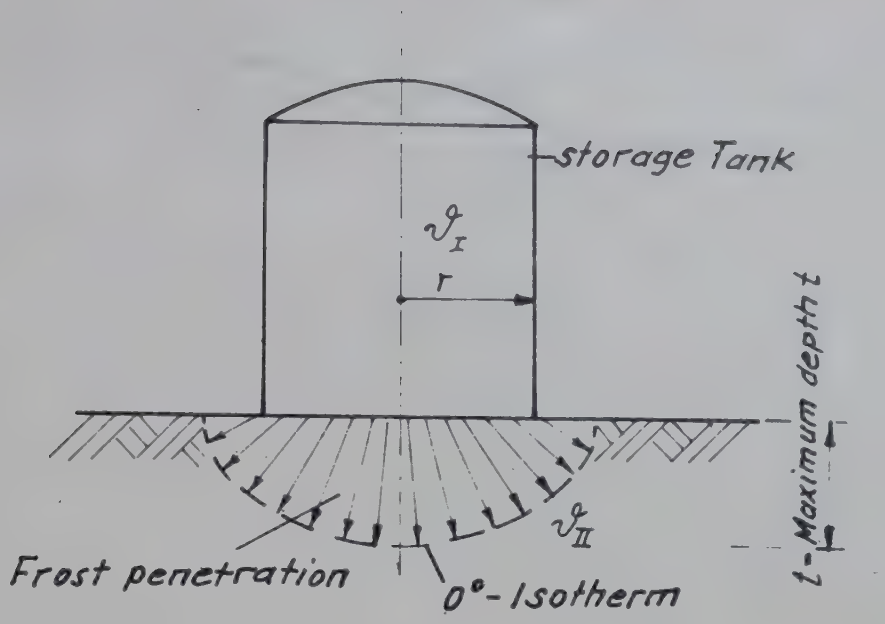

storage Tank
ϑ_I
r
t - Maximum depth t
ϑ_{II}
Frost penetration
0° - Isotherm

Fig. 1

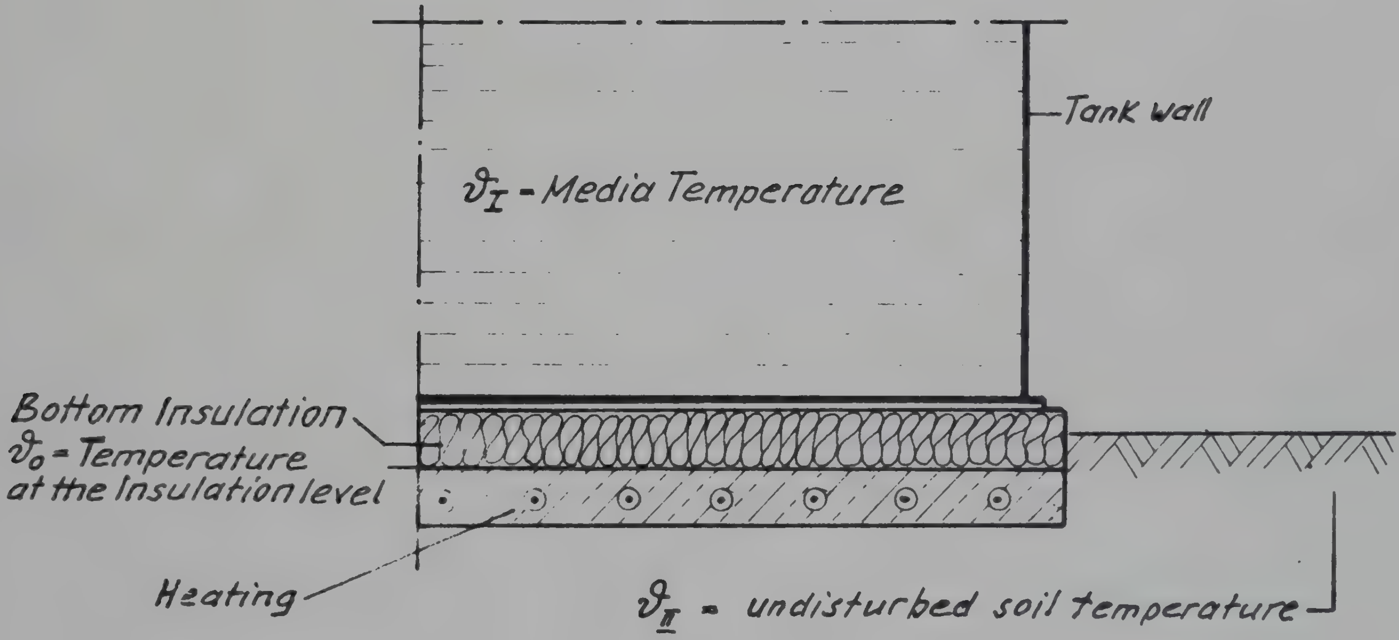
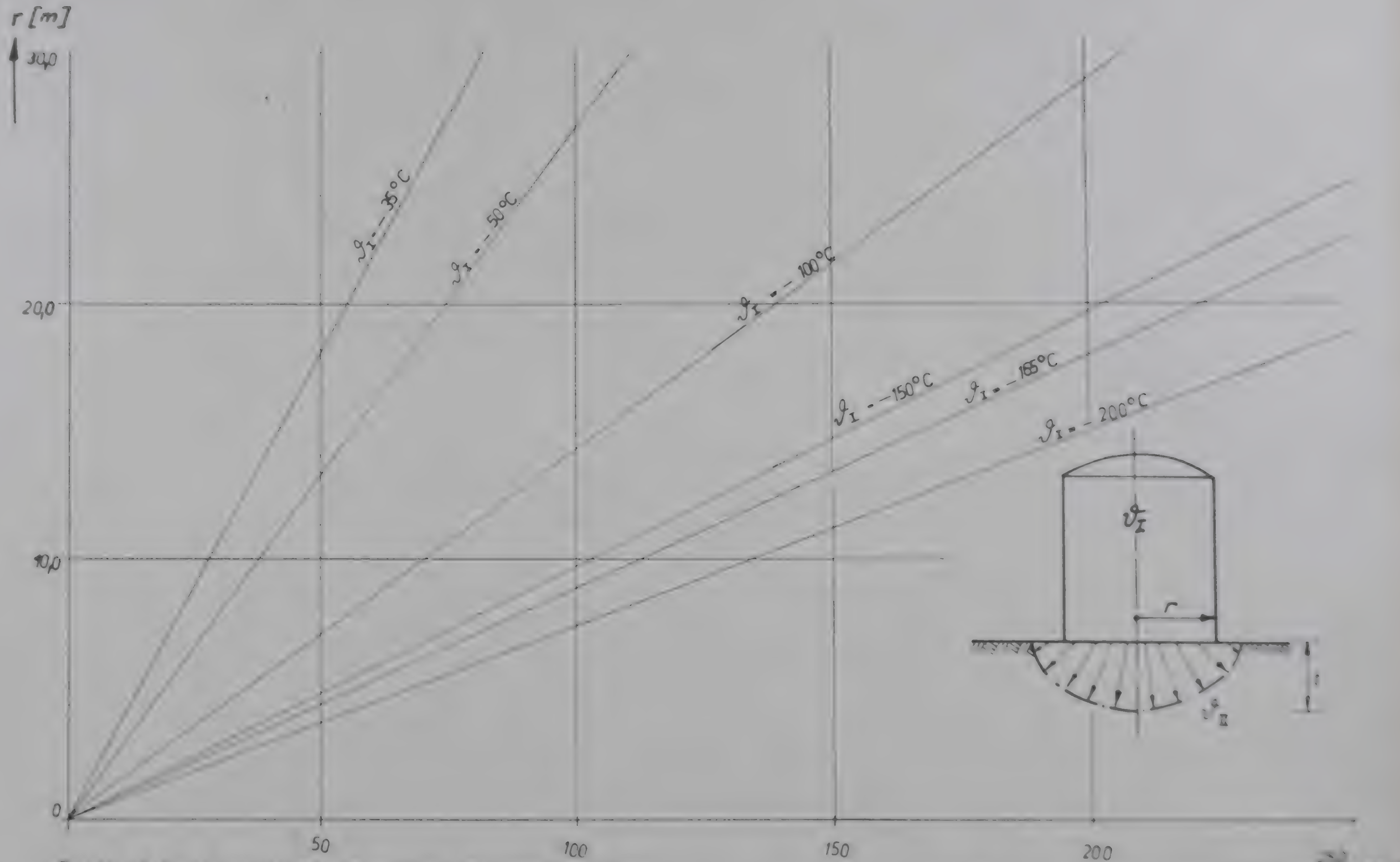
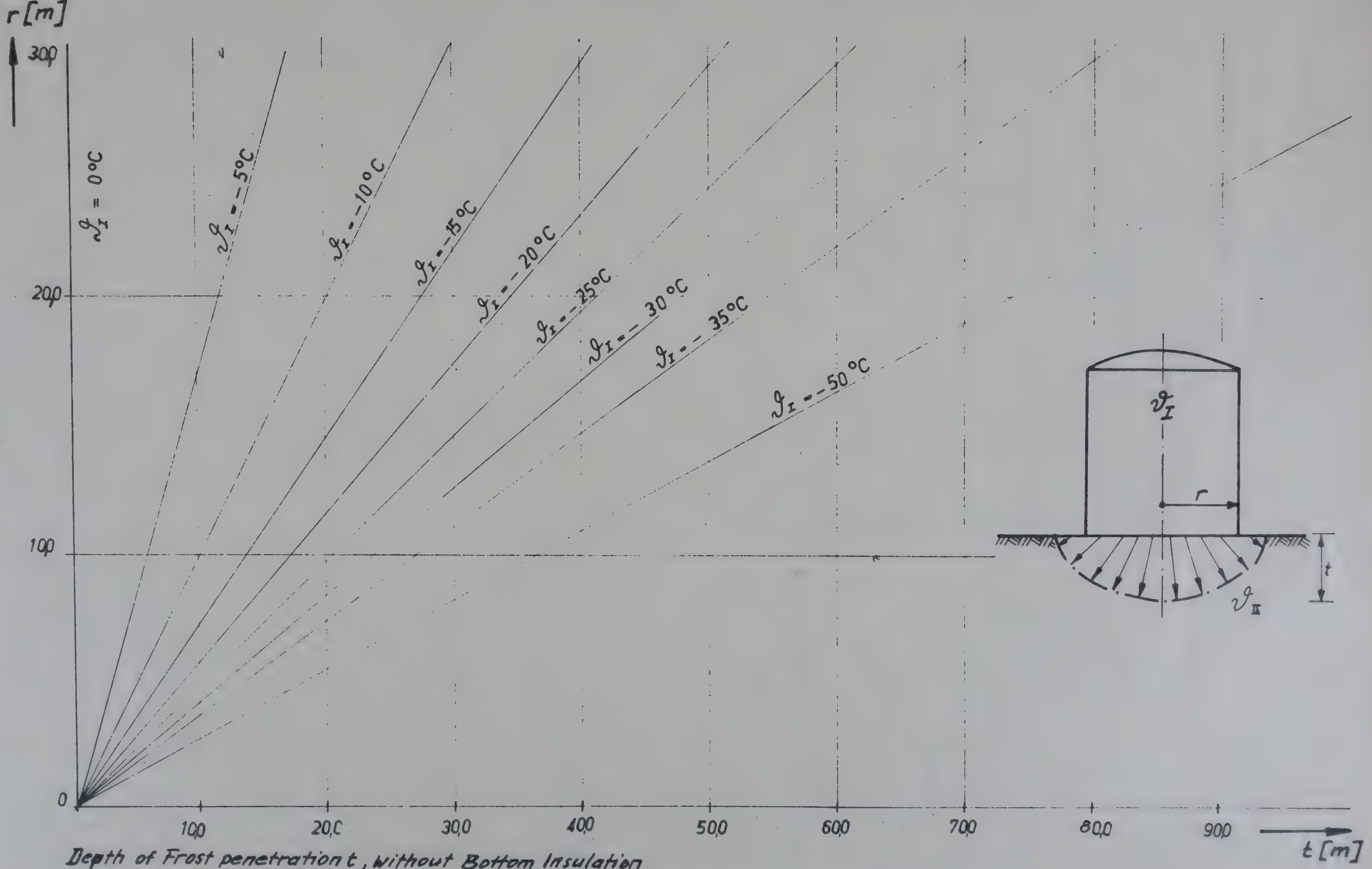

Tank wall
ϑ_I - Media Temperature
Bottom Insulation
ϑ_0 = Temperature at the Insulation level
Heating
ϑ_{II} = undisturbed soil temperature

Fig. 5



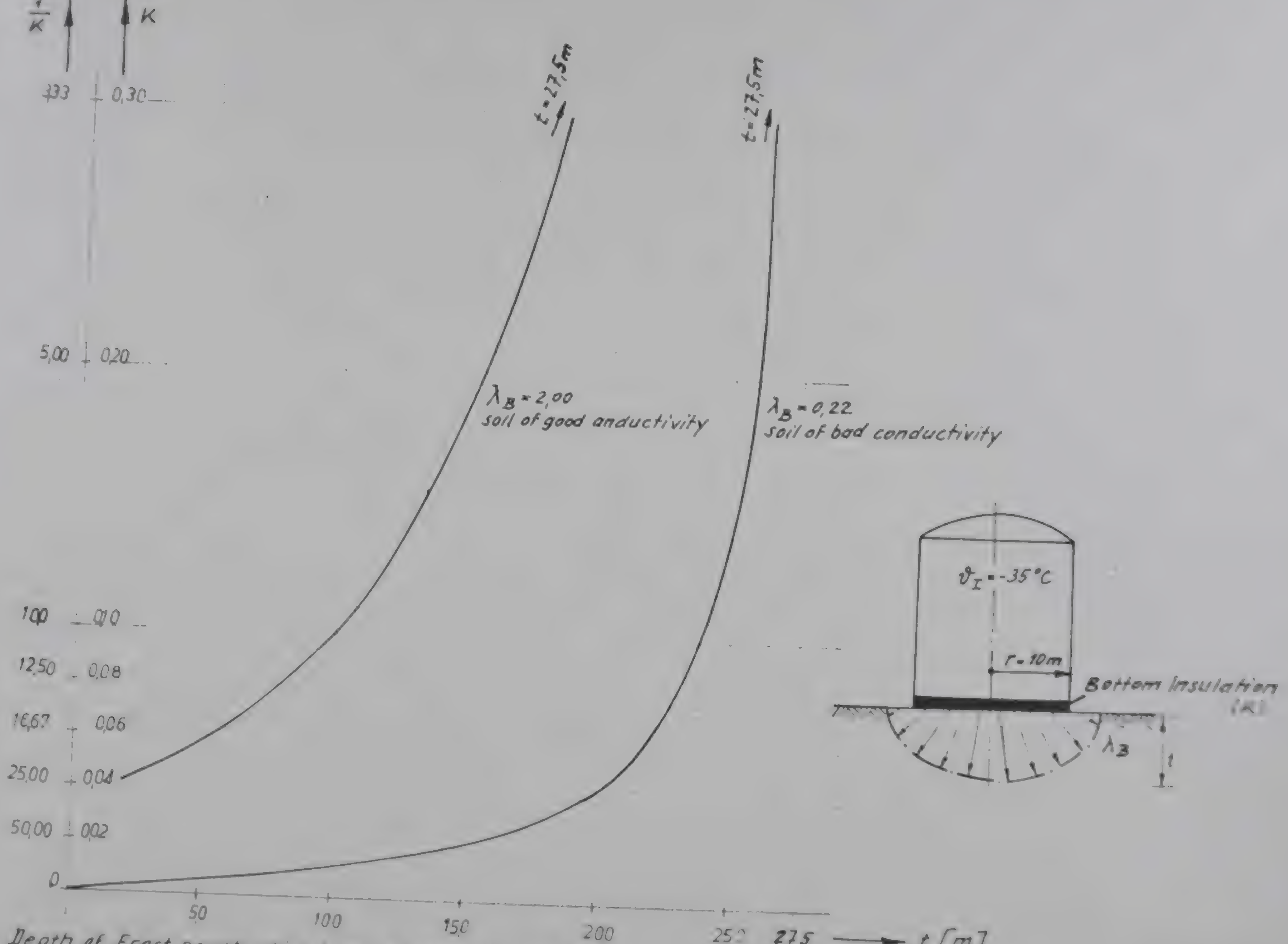
Depth of Frost penetration t, without Bottom Insulation,
as a Function of Tank radius r and the temperature of the Tank media ϑ_I = − 35°C to − 200°C

Fig. 2

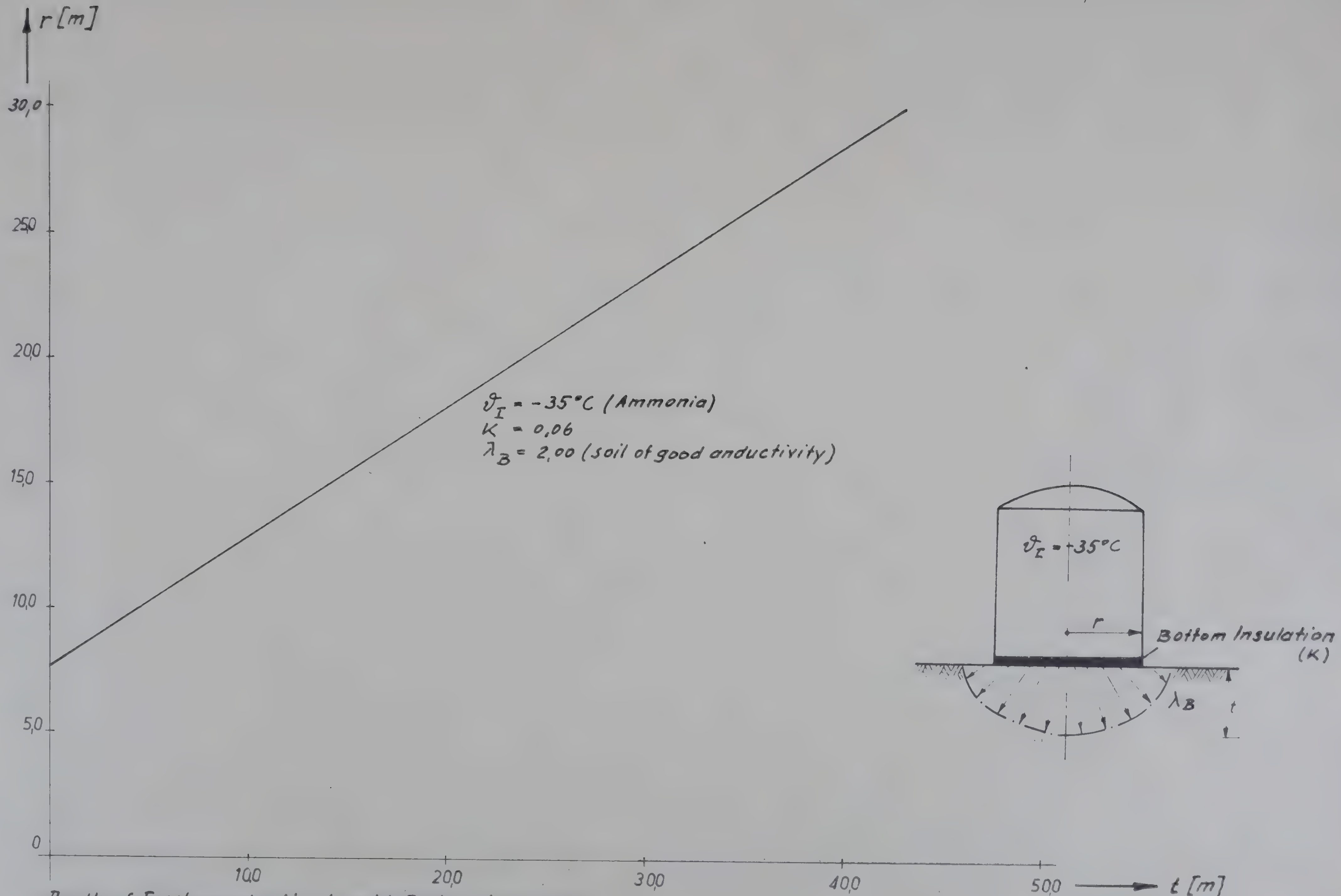


Depth of Frost penetration t, without Bottom Insulation, as a Function of Tank radius r and the temperature of the Tank media $\vartheta_I = 0°C$ to $-50°C$

Fig. 2a

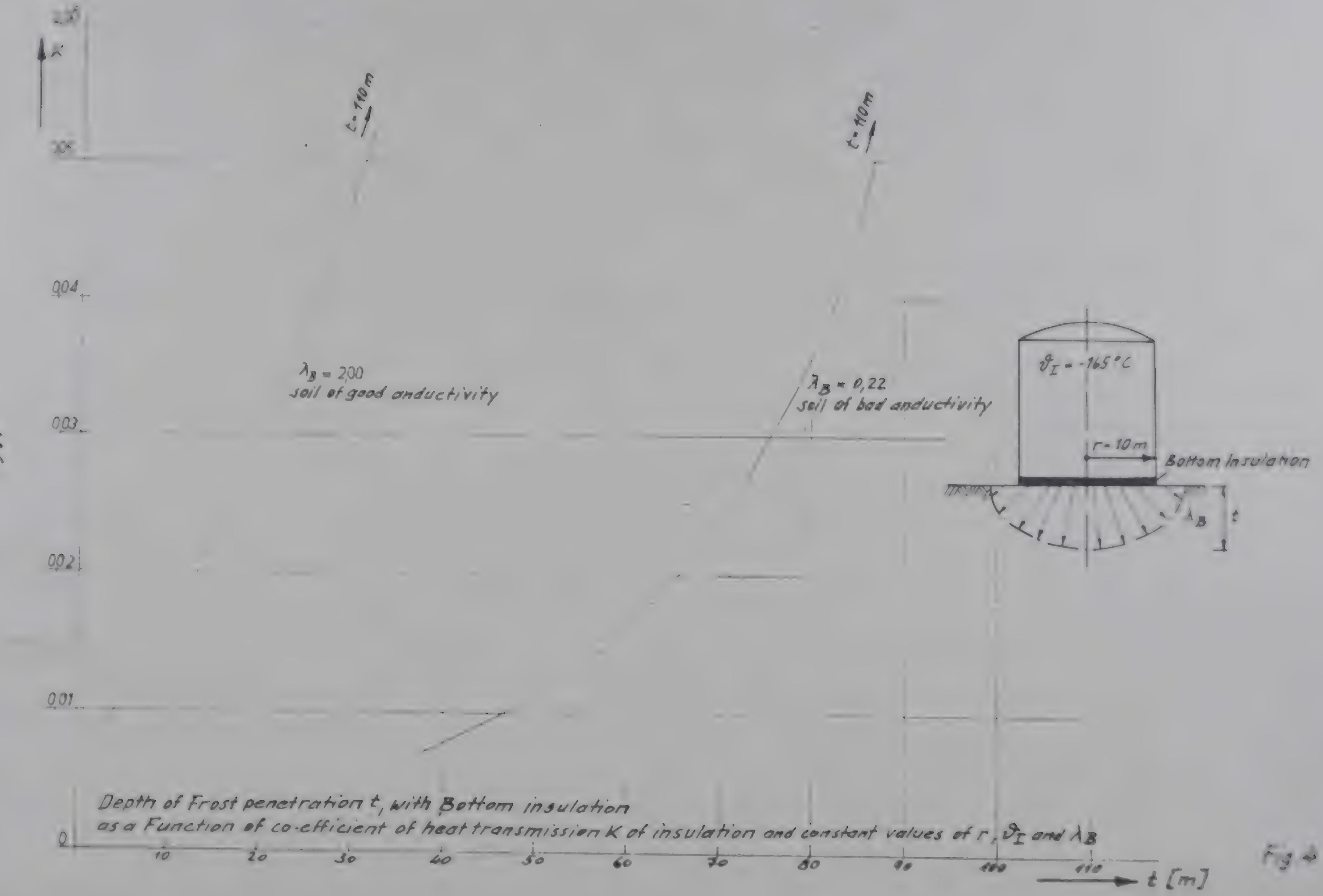


Depth of Frost penetration t, with Bottom Insulation,
as a Function of co-efficient of heat transmission K of Insulation and constant values of r, ϑ_I and λ_B Fig. 3



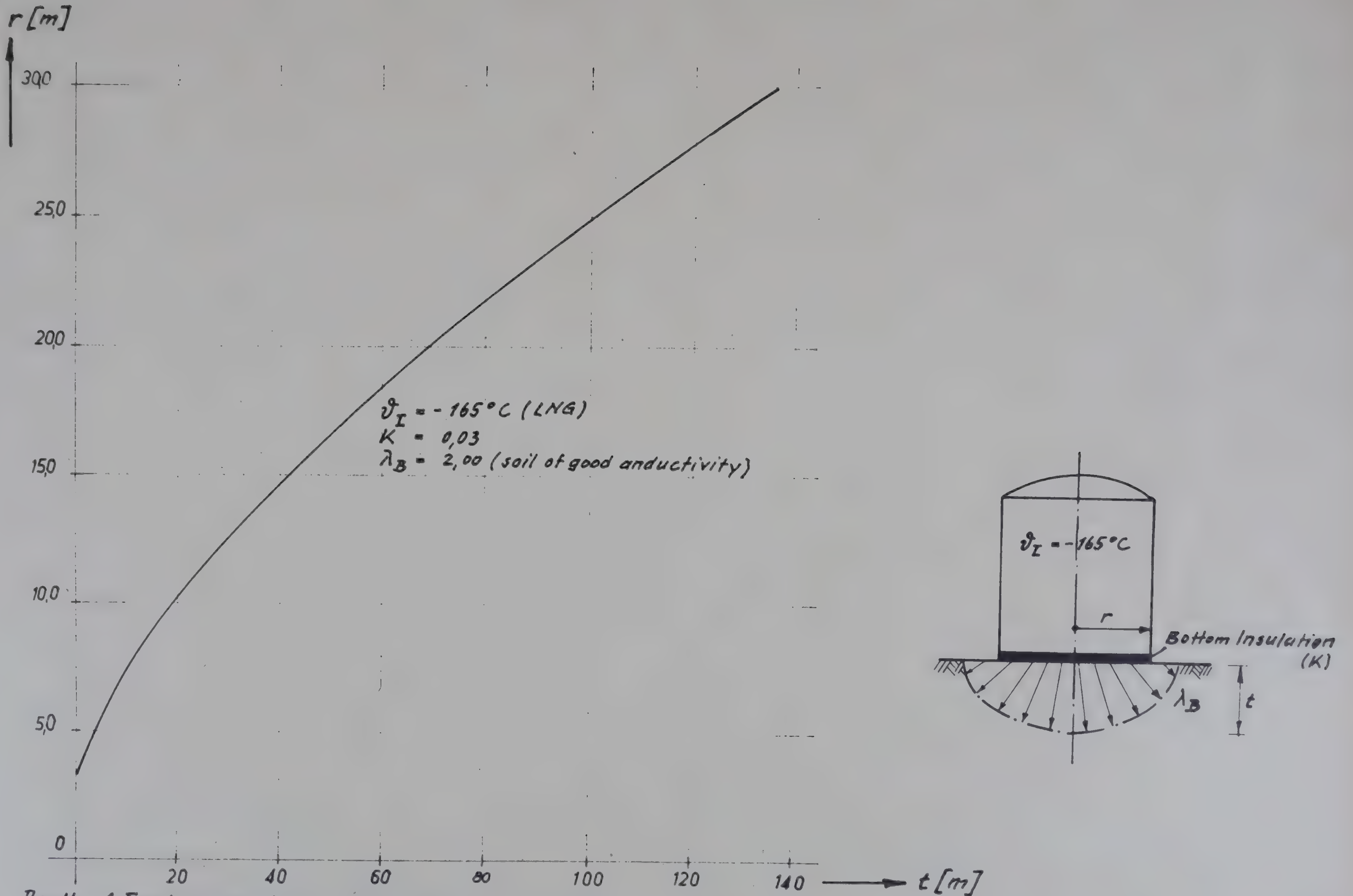
Depth of Frost penetration t, with Bottom Insulation, as a Function of Tank radius r and constant values of ϑ_I, K and λ_B

Fig. 3a



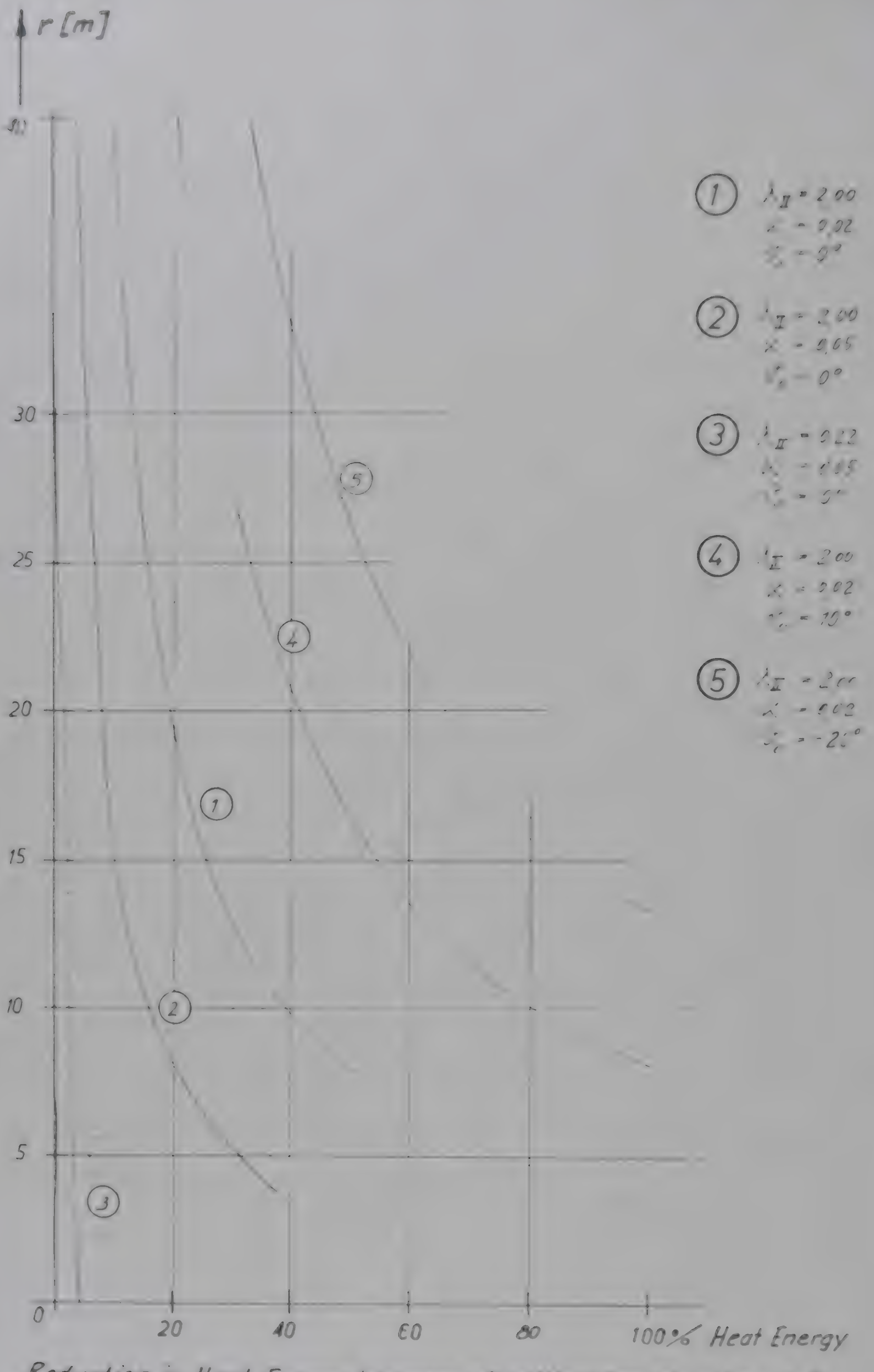
Depth of Frost penetration t, with Bottom insulation
as a Function of co-efficient of heat transmission K of insulation and constant values of r, ϑ_I and λ_B

Fig 4



Depth of Frost penetration t, with Bottom insulation, as a Function of Tank radius r and constant values of ϑ_I, K and λ_B

Fig. 4a



Reduction in Heat Energy because of utilisation of soil heat ϑ_{II} as a Function of Tank radius r for $\vartheta_I = -165°C$ and $\vartheta_{II} = +10°C$ (100 % Heat Energy = required Heat Energy without soil heat)

Fig 6

LIQUEFIED NATURAL GAS STORAGE AT AMBERGATE, DERBYSHIRE, ENGLAND

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INTRODUCTION

The L.N.G. storage facility at Ambergate is in course of construction by Messrs. Whessoe Ltd., Darlington. At the time of preparing this paper the tank is complete. Materials and equipment associated with the ancillary plant are available, and site construction proceeds towards an expected completion date of late Spring 1969.

Plant associated with the 5000 tonnes capacity storage vessel includes a reliquifier for handling the boil-off and is located about 50 metres from the tank. It is the first plant of its kind in the U.K. gas industry and ultimate operation should provide useful data for this type of plant.

The remaining equipment consists mainly of the three Brown-Bingham L.N.G. discharge pumps and associated vaporisation equipment, together with a road-tanker off-loading/loading bay, the former duty for use during the annual main tank refill. It is planned to road-haul L.N.G. from the Gas Council Canvey installation. Conversely, the loading of road tankers may become a necessity within the Board's Area, bearing in mind future district conversion from manufactured to natural gas. A smaller L.N.G. pump, the prime duty of which will be to recirculate L.N.G. in order to prevent stratification of the main tank contents, will be utilised for such tanker loading. Road tanker off-loading will be accomplished by pressurisation as mentioned later in this paper.

The main discharge pumps will be located inside an earth bund surrounding the tank; the refrigeration equipment and a steam boiler installation, providing heat for L.N.G. vaporisation, are both located outside this bund in adjacent brick-constructed houses.

GENERAL

The Ambergate towns-gas manufacturing station is designed to produce 100 million ft^3/day (2.8×10^6 m3) utilising natural gas for reforming, burning, and enrichment. The demand for natural gas under these conditions is 60 million ft^3/day, (1.7×10^6 m^3/day) equivalent to about 1110 tonnes.

Alternatively, naptha may be used for reforming and burning, leaving natural gas simply as an enriching agent. Under these conditions the natural gas requirement is about 600 tonnes/day and this is the proposed method of operation.

The choice of natural gas vis-a-vis L.P.G. as an enricher followed from the projected increase in availability of natural gas; the proximity of the Ambergate site to the national methane pipeline additionally suggested that natural gas was a logical material for enrichment. The utilisation of, and necessary storage facilities for, the L.P.G. alternative would have represented a capital investment of what appeared to be, and continues to appear, of short term value only.

The decision to use natural gas posed the problem, however, of interruption in town-gas production in the event of a shortage of natural gas or, in the extreme, a total failure in supply. It was to cover this eventuality that the installation of an L.N.G. storage facility was largely decided, although additional benefits were evident :

(a) The L.N.G. installation would assist in meeting peak demands for natural gas as conversion proceeded.

(b) A reduction in diurnal load variations on both town gas and natural gas systems would be possible.

As mentioned earlier the Ambergate gas production plant requires 600 tonnes of L.N.G. per day for enrichment only when operating at nominal output and it was considered prudent to instal a storage facility of 5000 tonnes, representing about 8 days operation.

The mode of storage for such a quantity was clearly in favour of the conventional insulated double-skin above-ground tank since neither the size nor the local soil formation made frozen ground storage a viable proposition.

The flow diagram shows the proposed connections relating to the L.N.G. facility and the gas production plant. It will be seen that locally stored natural gas used on the production plant is first delivered to a metering system and is under the control of Gas Council. Thus the storage does not represent an exclusive source of feedstock for the adjacent reforming plant.

The Tank

The design of the tank at Ambergate follows closely those constructed at Gas Council's Canvey installation during 1958 and 1963 and which have been described by Walters & Ward. Similar tanks constructed currently in the U.S.A. and also since that time in other parts of the world have been described elsewhere.

The Ambergate tank rests on a concrete raft supported on 225 plinths, each a continuation of a bored pile. These plinths afford a 0.75 m. air space which will prevent frost-heave. The photograph attached illustrates the hilly nature of the site, and notwithstanding an initial earth and rock-moving operation of considerable magnitude, the levelled site remained one consisting of massive faulted grit-stone interspersed with shales. The piles varied in length from 3.5 to 5 metres consistent with the steeply dipping nature of these strata. It was necessary to plan carefully all facets of construction, including the removal of spoil, having regard to the complexity of building operations associated with the adjacent reforming plant which was currently in hand. It is interesting to note that the total rock and other spoil removed from this entire site, which covers only some 44 acres (17.8 hectare) approached 10^6 metre^3 (say, 1½ million tonnes). Not least of the problems was transport of this material from site and the location of other sites for its disposal.

The mild-steel outer tank was first constructed since it would afford a relatively dry environment for the subsequent construction of the inner L.N.G. container. It is 0.635 cm thick 28/32 tonnes steel conforming in design to BS. 2654. All material conforms to BS.15 with the exception of the annular base ring which is 0.795 cm. thick and to BS.1501 - 224 LT.15. Five external stiffeners are attached to the shell which is 30.6 m in diameter by 20.8 m straight side. The spherical roof has a radius of 46 m.

The base of this tank is covered with layers of rigid load bearing "Foamglas" insulation of approximately 51 cm. total thickness bedded on a 5.1 cm. thick screed and topped with 6.3 cm thick concrete slabs.

The inner tank is designed generally to APT.620 from NP.8 -OA1/Mg. alloy (203-5083) and is 29 m. diameter by 19 m. wall height. The rafter supported roof is of spherical form and gives a total height of 22 m. measured from the flat base. At the time of the finalisation of design of the tank, the Middle East situation was such that importation of L.N.G. into Canvey Island was interrupted, and it was therefore considered desirable to design the tank to store L.P.G., specific gravity 0.6, for this could be used as an alternative enrichment for the reformers. The tank is designed to withstand additionally an internal gas pressure of 100 mm Hg, or alternatively, a vacuum of 5.6 mm Hg.

The inner tankis of course prevented from lifting due to internal gas pressure by means of the usual holding-down bolts which pass through the base insulation and the concrete raft. Anti-rotation pins and runners are also provided.

The diameters of the inner and outer tanks provide a 0.63 m. annulus, Perlite filled. This interspace will be maintained under a small pressure of dry nitrogen to ensure an inert and moisture free atmosphere.

The inspection of both inner and outer tanks, prior to and during construction, has been comprehensive. Examination of plate preparation, alignment and gaps of joints, welding examination and cleaning, has been in line with the high standards laid down for this class of work.

Upon completion of construction the inner tank was hydraulically tested to a depth of 14.5 metres and the gas space pressurised to prove the adequacy of the relief valves and the tightness of the roof. The annulus was also air tested to a pressure of 18.7 mm Hg.

Connections welded to the inner tank are N8 alloy. Two connections, a 25.4 cm diameter L.N.G. outlet and a 5.1 cm. diam. L.N.G. overflow, are butt welded into the shell near the bottom. All other attachments, including the fill line, are via the 61 cm. diameter N.8 central riser located centrally in the roof; a S.S. bellows traverses the interspace between the L.N.G. container and the shell. The relief valves are attached to the riser terminal as shown in the photograph and consist of a pair of pilot operated valves set at 77 mm. Hg. plus three 20 cm. bore emergency vents set at 94 mm. Hg, these latter are for fire exposure duty.

Consideration was given to the possibility of ignition of vented gas and the effect of this heat release upon the adjacent aluminium riser. It was thought advisable to extend the relief valve outlets by means of distance pieces. Additionally, the dome of the riser will be insulated.

Each relief valve is preceeded by a block valve, mainly as an aid to maintenance and adjustment. The precaution has been taken, however, of inter-locking these valves so that only one valve out of the five can be closed at any one time, the remaining four being capable of dealing with discharge under the worst conditions. A further aid in the event of fire will be a small access platform located between the relief valves and their attendant block valves. This platform will be covered with 5.1 cm. of refractory concrete and will act as a barrier to radiant heat.

BOIL-OFF DISPOSAL

Previous L.N.G. Storage Tanks installed for the Gas Council at Canvey Island have been associated with gasholders, which receive the boil-off from the tanks, and from which the gas is compressed into a local main before subsequent reforming. However, at Ambergate no such facility existed, and, as the plant was to be associated with both the reformers and the natural gas main, consideration was given to the possibility of compressing pre-warmed boil-off gas into either of these systems.

This investigation proved that the compression of boil-off gas would be more economical than reliquefying the gases, but if adopted would promote the undesirable features of increasing the specific gravity of the tank liquid throughout the period of storage, and injecting gas into the natural gas grid main at a higher calorific value and density than the existing contents. Additionally, without reliquefaction of boil-off gases from the tank, some topping-up of the tank would be required to maintain a full tank for use during peak periods.

Taking into consideration the heat gains into the system through the tank walls, refrigeration lines, unloading of filling tanker etc., enquiries were issued for reliquefaction equipment to deal with a total heat input of 55kW, or an evaporation rate of methane equivalent to 860 lb per hour (390 kg/h).

The plant ultimately incorporated into the contract was that offered by Sulzer Bros. of Switzerland, and this includes helium compressors, heat exchangers and expansion turbines in a cold box, and condensers.

The working fluid, helium, is compressed at ambient temperature to a pressure of 12 atmospheres before passing through a heat exchanger counter-current to cold helium returning from an expansion turbine. This high pressure, low temperature gas at 92°K passes through the cold line to the methane gas condensers mounted on top of the storage tank. This warmed helium gas returns to the cold box and passes through two expansion turbines in series reducing its pressure to 6 atmospheres before the heat exchange mentioned above. Here the helium is returned to ambient temperature for subsequent recompression.

Normal maximum operating conditions require the use of one helium compressor, two expansion turbines and one condenser, but one additional unit of each is included in the arrangement to provide the necessary degree of standby.

The compressors are oil-free in operation, are single stage two cylinder machines hermetically sealed to minimise helium losses, and the turbines are single stage centripetal units, arranged vertically with an oil brake at the lower end to absorb work output. Speed of operation of the turbines is 85000 rpm.

The coefficient of performance is 12% which represents an input of 458 kW for the required 55kW output.

VAPORISATION

The steam heated vaporisers have been installed to provide a total required capacity of 60 tonnes/hour. Each of these units consists of a horizontal M.S. single shell containing a four pass S.S. tube bundle through which liquid methane passes. As steam is admitted to the shell at an uncontrolled rate, the vaporised methane temperature approaches that of the 100 lb f/in^2 (7 atmospheres) steam, and desuperheating is obtained by injection of L.N.G. immediately after the vaporiser. The quantity of L.N.G. so admitted is 8 tonnes/hour, and the L.N.G. through the tube bundle is 24 tonnes per hour.

This type of vaporiser was adopted following full investigation into alternative types such as submerged combustion heaters, indirect type vaporisers utilising intermediate fluid etc.

At Canvey Island, water heated, steam heated and indirect type of vaporisers had previously been installed and operated, but at Ambergate no plentiful supply of water or space existed and these factors limited the ultimate choice.

Water heating naturally could not be adopted, and even though submerged combustion heaters appeared to have lower installation costs, the space available to position these safely in relation to the storage area was not in their favour. At that time also, no first hand operating experience in this country was available; the proposed units at Canvey Island were not then working, and it was generally considered desirable that the use of this type of vaporiser for a peak load installation should follow experience of operation and determination of maintenance and running costs.

Considerable attention was given to the incorporation of vaporisers using isopentane as the intermediate fluid. In this method, the isopentane is vaporised by steam or combustion products from direct-fired equipment, and this in turn imparts its heat to the L.N.G. via a tube bundle. However, when compared with the simpler direct steam vaporisers , this method was more expensive.

By the utilization of stainless steel tubes in the bundles and ensuring adequate supply of steam to the shell at all times, the anticipated disadvantages of high temperature gradient across the tube walls and the freezing of steam condensate in the cheaper and simpler type of vaporiser are expected to be avoided.

PIPEWORK

All pipework handling L.N.G., together with the cold helium pipework, is fabricated from stainless steel to ASTM. 312-304. They are schedule 40 and 10 respectively. The choice of this material follows the present-day trend in selection and had regard to its strength, ductility and dimensional stability at low temperatures. Its higher melting point compared with Mg/Al alloy has much to commend it when considering possible exposure to fire conditions. An additional bonus stems from the relative ease of welding this material, and when used in conjunction with stainless steel flanged valves bolting problems are considerably reduced.

Insulation of L.N.G. piping will be by preformed polyurethane with metallic cladding. The cold helium lines between the cold box and the tank are carried in a mild steel duct, Perlite filled, connected to the base of the tank. The helium pipes pass into the tank interspace and continue upwards through the Perlite insulation, terminating at the condensers which are housed in an insulated mild steel box located at a point on the periphery of the tank roof.

VALVES

In the early days of the liquid methane importation project at Canvey Island it was decided to employ, for economic and supply reasons, gunmetal valves for all cold duties, and this gave rise to the adoption of two types of flange bolting to compensate for differential contraction in the mains, which again, for economic reasons, were of aluminium.

Following exhaustive investigation, it was found that valves provided by Messrs. Walworth-Aloyco offered advantages over those previously used for L.N.G. duties at Canvey Island. The wedge is specially designed to contain a slot or groove to provide flexibility in closing and this with double sealing rings of 'KEL-F' provides practically bubble tight closing. Tests were carried out, in this country and abroad, using liquid nitrogen and methane to test alternative types of valves, and upon the results of these tests, it was finally decided to import these valves from Milan.

These valves have bodies generally of cast A.312(CF 8M) with F.304 trim. Gate valves have been specified where the superior flow characteristics are important, and vary in size from 5.1 to 25.4 cm diameter. Some globe valves will be used, mainly for control valve by-pass duty.

Representative valves of those manufactured for Ambergate were selected and tested under liquid nitrogen conditions, pressurised to 15 atmospheres on the upstream side after being closed by the application of a measured torque. Such tests were of course preceeded by hydraulic tests on all the valves to be used. A photograph of the acceptance tests arrangement in Milan of some of the valves manufactured for Ambergate is included in this paper.

L.N.G. TRANSPORT

The sole source of L.N.G. at present in this country is situated at Canvey Island, and it is from here that the liquid will be brought for the cool-down of the completed tank and equipment.

Road transport costs were assessed for the transportation over the distance of 180 miles (290 km), and these were significantly less than those associated with the installation of a liquefaction plant of a size suitable to fill the 5000 tonnes capacity tank.

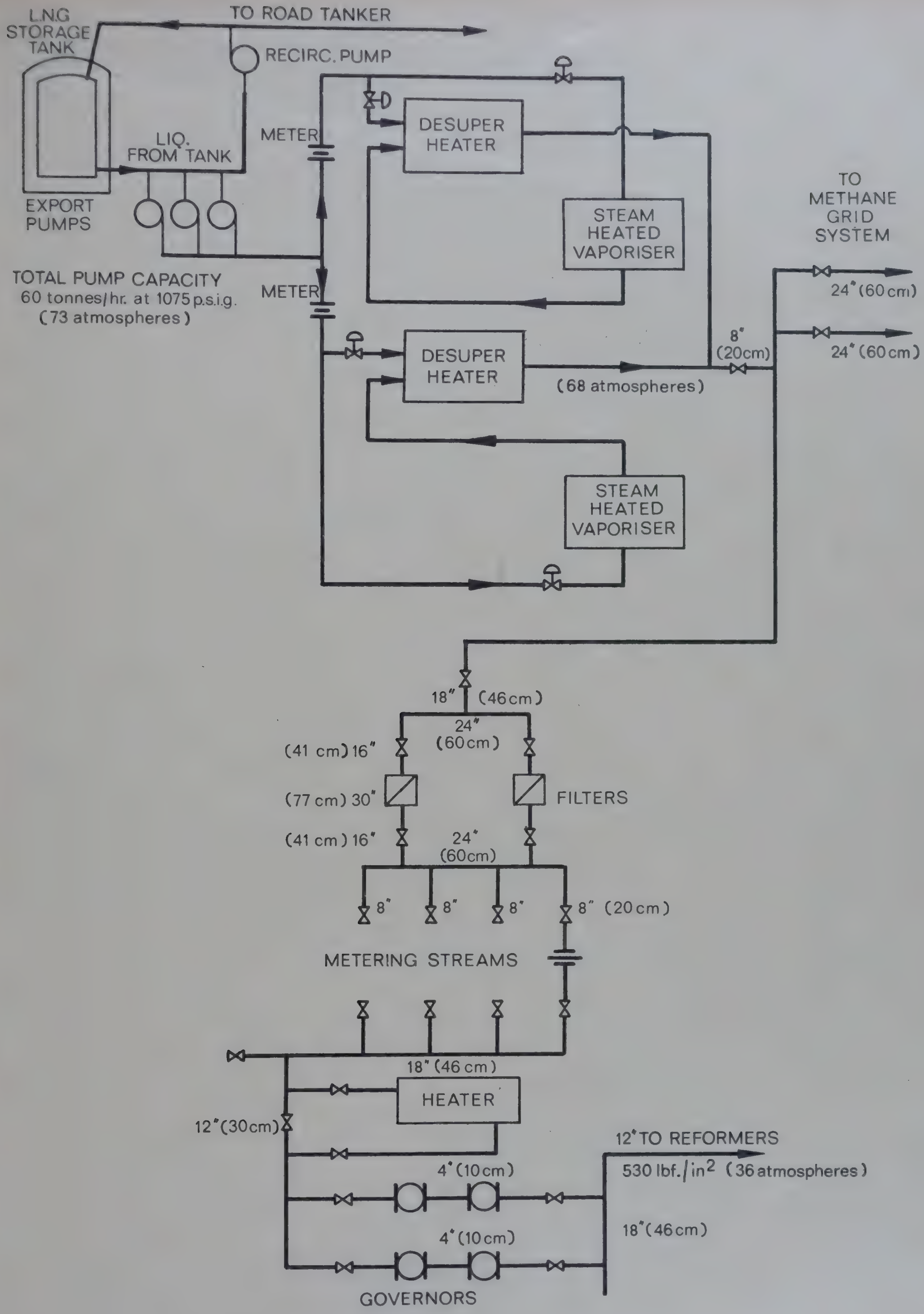
Arrangements have been made to hire initially two tankers, capacity 14½ tonnes, made from 9% nickel steel, and insulated externally with expanded polyurethane. Each tanker will do one round trip per day.

The unloading of the tankers at Ambergate will be achieved by pressurising the gas space in the top of the tanker to about 30 lb f/in^2 (2 atmos.) using vaporised L.N.G. Utilising methane gas from the distribution system at higher pressure, appeared to offer advantages, but as the gas has been previously odorised, and could involve difficulties associated with the deposition of water etc., this method was discarded, as also was the idea of using high pressure nitrogen, due to its solubility in liquid methane.

During the initial cool-down period, L.N.G. will be slowly fed into the tank, and the resulting boil-off gas will be reliquefied in the plant, which will have been previously cooled.

CONCLUSION:

This paper has been prepared before completion of construction and commissioning, and effort has been made to indicate some of the investigations and points that have arisen.


L.N.G STORAGE TANK
TO ROAD TANKER
RECIRC. PUMP
METER
DESUPER HEATER
LIQ. FROM TANK
STEAM HEATED VAPORISER
EXPORT PUMPS
TO METHANE GRID SYSTEM
TOTAL PUMP CAPACITY
60 tonnes/hr. at 1075 p.s.i.g.
(73 atmospheres)
METER
24" (60 cm)
8" (20cm)
24" (60 cm)
DESUPER HEATER
(68 atmospheres)
STEAM HEATED VAPORISER
18" (46cm)
24" (60cm)
(41 cm) 16"
(77 cm) 30"
FILTERS
(41 cm) 16"
24" (60cm)
8"
8"
8"
8" (20 cm)
METERING STREAMS
18" (46 cm)
HEATER
12" (30cm)
12" TO REFORMERS
530 lbf./in2 (36 atmospheres)
4" (10 cm)
18" (46 cm)
4" (10 cm)
GOVERNORS

Flow diagram - Ambergate L.N.G.storage

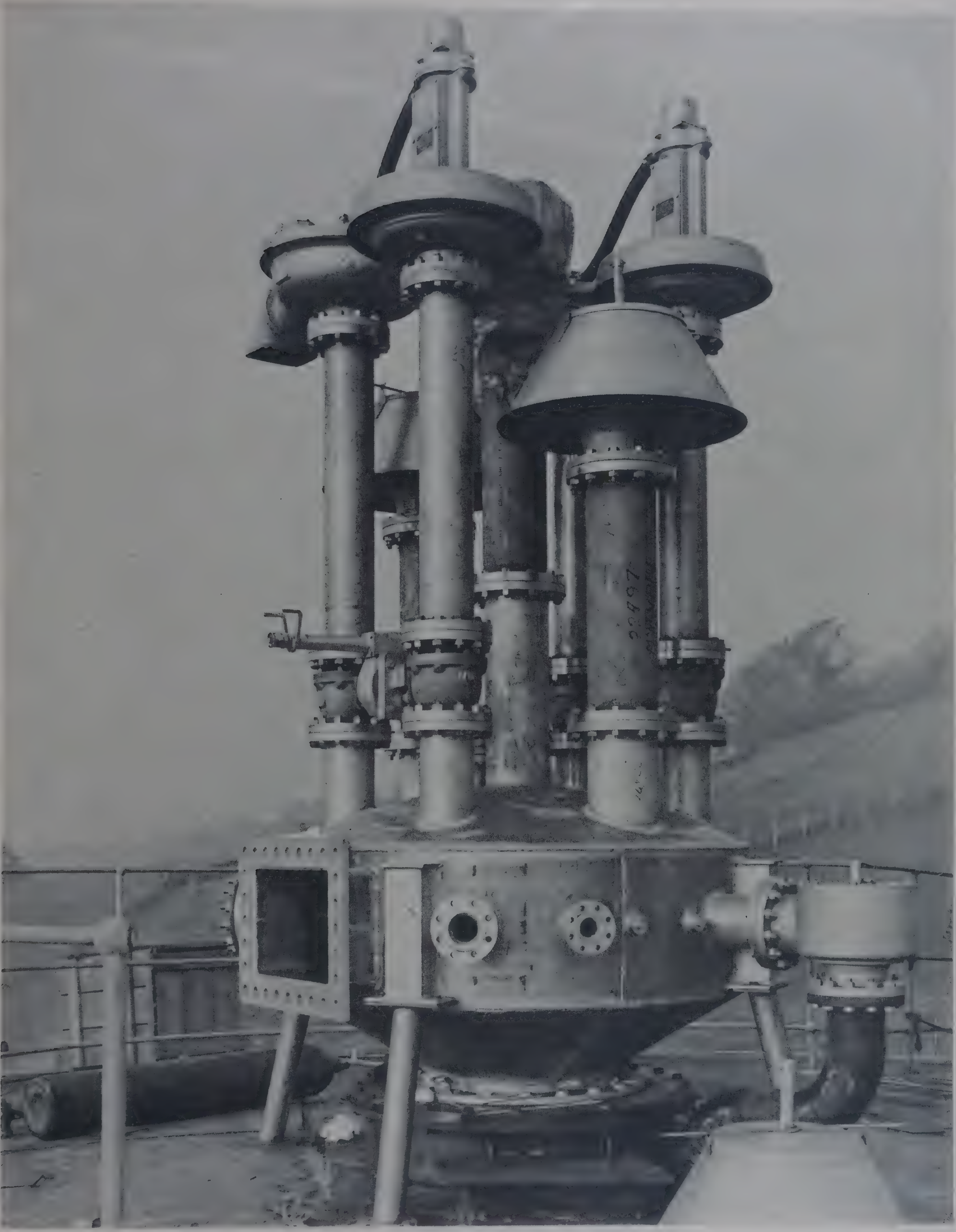


Fig.3. L.N.G.tank riser terminal

Storage of L.N.G. in Metallic Containers

by

J.K.S. Wardale

I INTRODUCTION

In view of the present interest in L.N.G. storage, a review of storage in metallic tanks of the double wall type is offered. Safety is a prime importance where the unit size is increasing and where economics might suggest the use of undesirable features. How far can one increase the size without going beyond a safe extrapolation from existing L.N.G. designs and experience from conventional oil tanks. What safety considerations are there in adopting higher roof design pressures for L.N.G. tanks now under consideration. Is there any foreseeable upper size limit?

II CURRENT SITUATION

L.N.G. is stored in concrete, frozen earth and metallic containers, principally the latter. This comes about because the L.N.G. storage vessel can be considered as a special form of oil tank, but nevertheless subject to the same general rules. Sizes can range from a few 1,000 barrels to upwards of 500,000 barrels, and can be built anywhere depending only on having good foundations.

API.620 Appendix Q (1) sets down standards of design, construction, inspection and testing applicable to L.N.G. service based on American practice. Other countries and international oil companies are also considering codes of good practice and a U.K. standard is under consideration. L.N.G. Technology is rapidly developing and the cost of storage is reducing all the time as for conventional storage with new design techniques, materials of construction and increases in unit size. 600,000 barrel tanks are now common. 1,000,000 barrel tanks are being constructed and tanks over this can be built when required.

On the other hand, if 600,000 barrels oil tanks can be built there seems little reason why similar L.N.G. tanks cannot also be built. Size alone therefore is not likely to prove a difficulty in the foreseeable future.

B.S. 2654 Part III (2) recently published, permits higher strength materials and higher design stresses required for such tanks.

III COMPARISON BETWEEN OIL AND L.N.G. TANKS

Although L.N.G. tanks can be considered a special form of oil tank, there are a number of important differences.

a) L.N.G. is normally stores at -160°C, its boiling point at atmospheric pressure.

b) L.N.G. storage is usually associated with liquefaction or a vapour handling system.

c) The Sp. G. of L.N.G. is around 0.45 compared with oil which varies between 0.7 and 1.0, and oil tanks are water filled on test compared with partial filling of L.N.G. tanks.

TYPES OF TANK FOR L.N.G. SERVICE COMPARED WITH OIL TANKS.

Oil is normally stored in

(i) Floating roof tanks

(ii) Freely vented fixed roof tanks - often with column supported roofs.

(iii) Pressure roof tanks.

(i) Floating roof Tanks: Used for storage of volatile products below their boiling point. Minimises evaporation from surface. Unless L.N.G. can be kept sub-cooled in storage floating roof tanks are not likely to be feasible.

(ii) Freely vented fixed roof tanks: Because of the low volatility of some oil products a weatherproof cover over the tank is all that is required. There are quite unsuitable for L.N.G. because of air contamination and sub-cooling required.

(iii) Pressure roof tanks: To date the fixed pressure roof tank has been universally adopted for L;N.G. storage. It prevents ingress of air, allows compressors to pull boil off and flash gas from the tank roof for subsequent processing, enables the tank to cope with atmospheric pressure fluctuations and provides a completely closed storage system.

A. STORAGE AT -160°C.

The low storage/of temperature has the largest single effect when comparing L.N.G. and oil storage. Oil storage design techniques can be applied to the liquid container but the low storage temperature necessitates special constructional materials. The tank must then be insulated to minimise heat leakage, prevent ice formation and in some localities prevent frost heaving of foundations.

B. VAPOUR HANDLING IN L.N.G. TANKS

However efficient the insulation, there will always be some heat leakage and boil off gas produced in storage. In addition the tank is often used as a vapour/liquid separator after the final pressure let-down stage in the liquefaction process. This makes it essential for the tank to be capable of operating with compressors which can draw vapour from the roof and where an operating range sufficient for control purposes can be accommodated within the roof design pressure.

C. WATER TESTING OF L.N.G. TANKS

Shell plate thicknesses are calculated from the product Sp. G. and the extent of water testing after construction is primarily one of economics. Water is generally the only liquid available for tank testing but a full water test would require thicker shell plates and stronger foundations than for L.N.G.

Current practice is to test the tank with about 25% more weight of water than the full product design load. This provides a 25% foundation overload and over-stresses the lower shell courses. A 25% overload applied to a design stress of $\frac{2}{3}$ proof stress for the shell plates equals 85% proof stress and is in line with vessel design practice.

Thus if anything of a structural nature was incorrect such a water test would almost certainly discover it with less likelihood of a disaster than if it occurred when the tank was filled with L.N.G. The higher strength of the tank when cooled to the storage temperature and the inherent ductility throughout the temperature range is an added safeguard. Water testing at ambient temperature therefore occurs when the vessel is in its weakest state.

In theory, therefore, an L.N.G. tank cannot be said to be fully tested until it is full of L.N.G. In practice partial water testing goes a long way towards ensuring satisfactory L.NG. service.

III DESIGN OF L.N.G. STORAGE TANKS

MATERIALS OF CONSTRUCTION

Materials of construction in common use are 9% Nickel steel and Aluminium alloys, with stainless steel as a possible material. These materials are ductile to at least -200°C. and their low temperature strength is higher than at room temperature. However under site conditions it is not always possible to match the plate strength with the weld metal strengths so that under conditions of stress intensification, for instance at shell connections, preferential straining may take place in the weld metal instead of in the plate. Correct design of critical areas can overcome this disadvantage which becomes more serious as the design stress for a specific material is raised. This is in contrast to oil tanks were the weld metal strength is normally stronger than the plate.

TANK DESIGN

Considering the tank as a liquid container considerable benefit is derived from conventional storage tank design techniques. Higher strength materials have been used in recent years and both in the U.S.A. and Europe design codes embodying fully proven materials and design techniques using higher design stresses have been published (2) (3) (4). The British standard also permits a design stress of $\frac{2}{3}$ the minimum yield strength up to a maximum of 37,500 p.s.i. and there seems little reason why a design stress of $\frac{2}{3}$ the 1% room temperature proof stress for cryogenic materials of construction should not also be used. A factor of 0.40 x U.T.S. may also be a desirable limitation until more experience with higher design stresses is available.

Oil tank design techniques also assist in specifying other structural features such as wind girders, shell stiffeners, bottom and bottom annular plates, the shell/bottom junction and shell connections.

Thus the experience already available from oil tank design and construction gives great assurance when considering the L.N.G. tanks now under consideration and although there are many other problems associated with the low storage temperature there is little doubt that proven design and construction techniques are available for the structural design.

ROOF DESIGN PRESSURE

It is in the general use of higher roof pressures that the structural design of cryogenic tanks differs from conventional fixed roof oil tanks. Few large oil tanks have roof pressures in excess of 8" w.g. and the majority are freely vented and use of holding down bolts is rare.

When the gas pressure is raised in a fixed roof tank containing liquid, a pressure is reached at which the shell starts lifting despite the weight of shell, roof and liquid holding it down. If the gas pressure is increased or the liquid level reduced, the amount of lift increases and vice-versa. If the bottom plate and shell to bottom junction is correctly designed some lifting can be tolerated but for cryogenic double wall tanks lifting is generally unacceptable because of the presence of the insulation and the effect of the lift on the design of through connections. Bolts attaching the shell to the foundations are therefore installed if at the design gas pressure and with an empty tank lifting will occur.

Theoretically therefore the design gas pressure can be quite high (several p.s.i.g.) provided enough holding down bolts are used and the roof is structurally adequate. However in these circumstances one must consider the behaviour of the tank in the event of gross over-pressurisation, it being assumed that the tank structure is adequate for the design conditions and remains so throughout its life.

Consider an example of roof over-pressurisation. An L.N.G. tank might have a roof design pressure of 1½ p.s.i.g and a product hydrostatic head varying from 5 to 20 p.s.i.g.

over the fully stressed portion of the tank shell.
If due to mal-operation the roof is over-pressurised, it might fail at 6 p.s.i.g., i.e. $4\frac{1}{2}$ p.s.i. above the design pressure. This amount of additional pressure is unlikely to lead to failure of the shell as it will have a factor of safety of at least 2 and the actual pressure would only rise from 5 to $9\frac{1}{2}$ p.s.i.g. at the top and from 20 to 24 at the bottom. If however the roof design pressure was $2\frac{1}{2}$ p.s.i.g., failure might be expected at 10 p.s.i.g. which could cause failure in the upper shell before the roof failed.

The same type of calculation can be done for the effect of over-pressurisation of the roof on the holding down bolts and it will be found that the holding down bolts are more susceptible to failure than the shell for any degree of over-pressurisation. This is particularly so when the liquid level is low and the holding down effect of the liquid reduced.

These examples show that the lower the roof design pressure, the more likely is over-pressurisation likely to cause roof failure before failure of either the holding down bolts or the tank shell. Every effort should therefore be made by the installation designer to keep the roof design pressure as low as possible consistent with operational efficiency.

While considering roof design, a comparison is useful between the behaviour of the conventional double wall design with that of the suspended roof design - see Figures 1, 2 and 3. In the suspended roof design, the outer roof takes the full gas pressure and therefore the gas pressure at which holding down bolts must be used now depends on the weight of the outer tank rather than the inner tank and there is no liquid hold down.

If over-pressurisation occurs, again the more likely point of failure is the holding down bolts although because the outer tank is always at ambient temperature additional robustness can be added to the bolts at small cost with the object of ensuring that a failure if it occurs takes place at the roof. The bolts can also be readily inspected.

However over-pressurisation will not occur provided the relief valves are fully effective and have capacity for normal or emergency conditions. Failures however have occurred due to over-pressurisation brought about by faulty relief systems and the likely behaviour of the tank should be known. (5), (6) (7).

FITTINGS AND CONNECTIONS

Like all oil tanks, a minimum number of fittings such as connections, access, level gauge is required and additional fittings specific to L.N.G. service are required such as thermocouples for critical temperature measurements, pressure gauges and pressure controller for maintaining the roof pressure at the desired value under continuous heat leakage and boil-off conditions. The double wall concept and low operating temperature also effect the detailed design.

a) Connections: It must always be remembered that the tank contracts both radially and vertically with fall in temperature and expands radially under the liquid head. See Fig. 4. Connections pass through the interspace between the tank and outer container and must be designed so that under all circumstances leakage is impossible because access to the interspace while the tank is in service is virtually impossible unless provision has been made at the design stage. It is also desirable to keep the number of connections which are below the liquid level to a minimum and wherever possible they should be positioned in the roof or upper shell rather than into the lower shell as is more normal for oil tanks. All low level connections should be robust with no joints in the interspace which could conceivably become a potential source of weakness.

b) Manholes: If possible all access should be from the roof.

c) Other Fittings: Many upper shell and roof connections are needed to provide a fully operational tank. As far as possible bolted connections should be avoided in the interspace but some relaxation is tolerable for roof connections. Many are also quite small and are associated with instrumentation. Here the important point is to ensure that instruments and associated isolating valves are accessible from the outer tank and can be kept ice free. The design of connections can follow oil tank design principles, due regard being paid to the materials of construction and the differential movement between the tank and outer container.

INSPECTION AND TESTING

Large L.N.G. tanks should be built to a high standard of design and workmanship and inspection and testing must be carried out on all critical components and fabrications as the work proceeds. However inspection and testing is a quality control tool to ensure that the actual fabrication will meet the standards assumed in the design. It should not be used illogically just because L.N.G. is a very cold and dangerous liquid. It is probably no more dangerous than many other liquids stored in bulk in a refinery or chemical plant complex.

It must be remembered that the materials are ductile at the service temperature. Design stresses are based on the minimum strength of the materials and assume good commercial standards of construction rather than absolute perfection. The extent of testing of components and inspection of fabrications should reflect this concept. For instance in a tank, the shell is only subject to a unidirectional hoop stress and many parts, for instance the circumferential seams, are unstressed. The degree of non-destructive testing should reflect this stress pattern. It is sometimes more difficult to weld thick plate to a given standard than thin plate and the degree of inspection could therefore be relative to thickness.

IV THERMAL INSULATION

L.N.G. tanks are insulated to prevent ice build up from contact with the atmosphere and to reduce heat leakage to a level consistent with the cost of handling the boil-off gas. The minimum aim of the designer is to prevent external frosting or condensation on the outside of the insulation in contact with the atmosphere and the double wall concept was developed as being the most economical and effective way of meeting this requirement.

Conventional oil tanks, vessels and pipework in cold service are often insulated with preformed slab or in-situ foamed material. However at -160°C the thickness becomes so great that it is difficult to support it effectively. Its effectiveness is less due to shrinkage of the insulating material which causes gaps to open up between slabs and increases water vapour diffusion and subsequent icing up. At the same time the temperature of the air or vapour in the interstices of the insulation varies with the operating level of liquid in the tank and the insulation 'breathes'. Such considerations effect any vessel or pipework carrying L.N.G. but L.N.G. tanks are so large, the insulation so inaccessible and periods between maintenance so long that anything as susceptible to the formation of cracks and fissures as conventional insulation could not be contemplated.

Thus once it is accepted that an external pressure tight envelope to contain the insulation is required, the use of powder insulation becomes feasible and economic and a wide range of suitable materials is available to fill the cavity.

BASE INSULATION:

The base insulation material must in the first place be a good insulant. It must also be capable of withstanding the weight of the tank and contents. In the event of leakage of L.N.G. into the bunded area and a subsequent fire there must be no possibility of fire damage which might cause it to lose its load bearing effectiveness.

SHELL AND ROOF INSULATION:

Powder and finely divided solids are suitable for filling the space between the tank and outer container, although any material which effectively prevents radiation and convection heat transfer between the inner and the outer tanks can be used. The voids between the particles are filled with purge gas at slightly over atmospheric pressure so that when the interspace 'breathes' there is no tendency to produce a partial vacuum. Nitrogen or natural gas vapour are the most common purge gases.

Some materials are better than others in preventing internal recirculation within the body of the insulation and it is important to recognise that some insulation materials show considerable variation between thermal conductivity under test and in service. This and heat leakage through connections between the inner and the outer walls, i.e. pipe connections, holding down bolts, instruments etc., must also be considered. The thermal conductivity of the purge gas would also marginally effect the heat leakage.

DIFFERENTIAL MOVEMENT BETWEEN TANK AND OUTER CONTAINER

Excessive external pressure may develop on the L.N.G. tank with dimensional changes in service when the interspace between the tank and outer container is filled with insulating powder.

Fig. 4 illustrates this effect which is due to both thermal expansion and contraction and elastic strain under the liquid head. Two solutions are to either retain the powder within a false wall inside which the tank can expand and contract at will or to ensure that the insulation system has some resilience to limit the external pressure to a tolerable level.

SUSPENDED ROOF DESIGN

Another roof design concept recently introduced known as the suspended roof is to build the tank without a pressure roof and instead to suspend a thermal barrier over the top of the open shell through which vapours can pass freely. The gas pressure is thus transferred to the outer tank but as the quantity of vapour required to pressurise the outer tank is small it is quickly heated in the space above the suspended roof and avoids low temperatures developing on the outer container. Figures 1, 2 and 3 illustrate the normal double roof construction and two forms of suspended roof. Heat leakage can remain the same because the thermal barrier can include sufficient insulation to keep it to what it would have been had there been a pressure roof.

The concept not only reduces the cost of the tank but allows the holding down bolts if required, to be transferred to the outer container where they can be inspected periodically. In fact all components subject to the roof design pressure are now transferred to the outer container and can be in mild steel. This then leaves the L.N.G. tank as a liquid container only.

CONCLUSIONS:

The object of this paper has been to show that double wall metallic L.N.G. tanks can be built larger than hitherto to give a safe and economical unit. Valuable experience can be derived from the design and construction of large oil storage tanks.

References:

(1) API.620 Appendix Q - Low Pressure Storage Tanks for L.N.G.

(2) BS.2654 Part III - Vertical Steel Welded Storage Tanks. (Higher Design Stresses)

(3) API.650 Appendix D - Welded Steel Oil Storage Tanks Alternative Design Basis.

(4) API.650 Appendix G - Welded Steel Oil Storage Tanks Higher Stress Design for Shells

(5) L.P.G. Tank Failure at Ras Tenura - Hydrocarbon Processing and Petroleum Refinery - November 1963.

(6) Failure of L.P.G. Pressure Storage at Fayzin - France. Fire International No. 12 - April 1966.

(7) Failure of Liquid Ethylene Storage Tank at I.C.I. - North Tees. To be published.

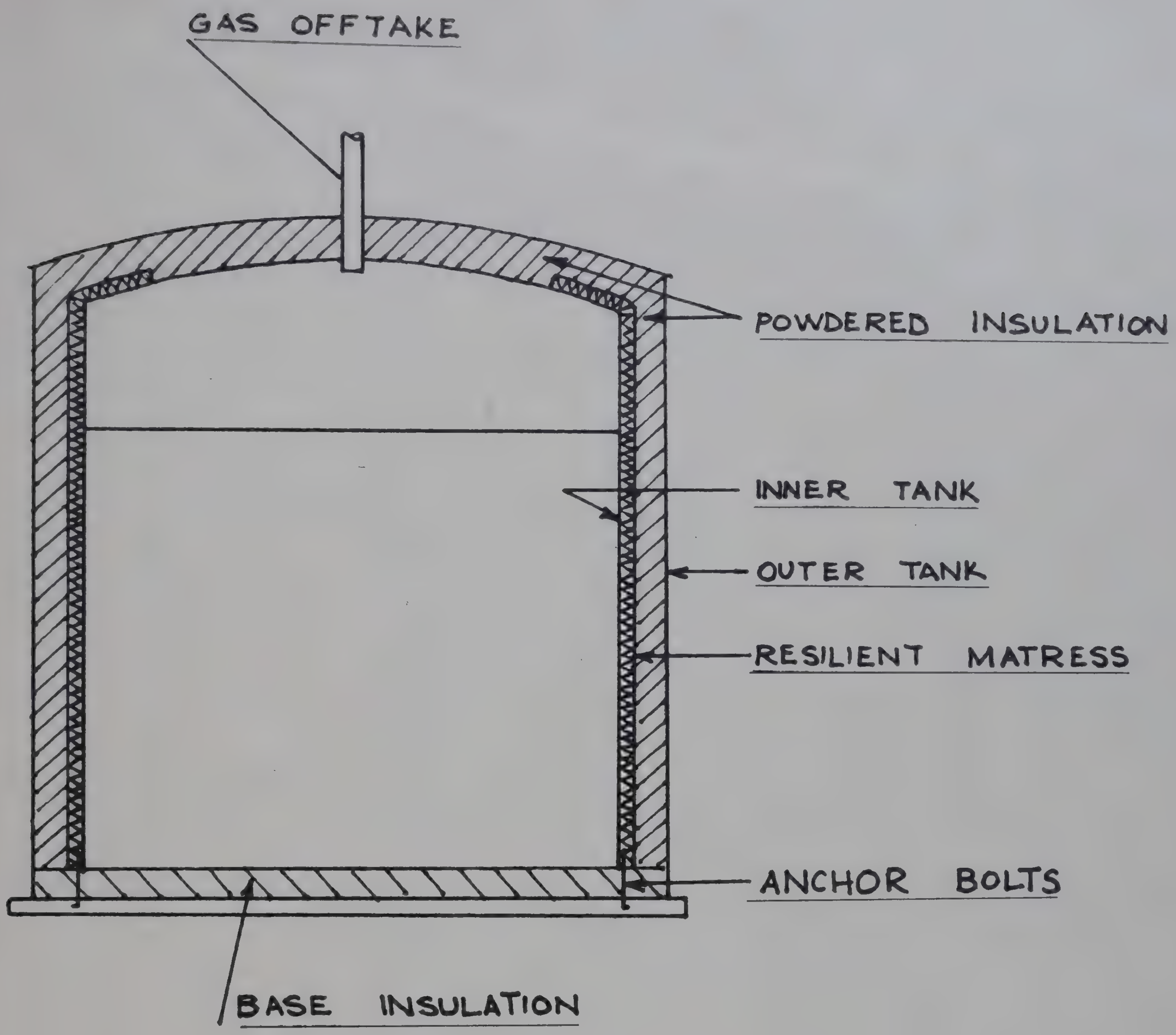

GAS OFFTAKE
POWDERED INSULATION
INNER TANK
OUTER TANK
RESILIENT MATRESS
ANCHOR BOLTS
BASE INSULATION

Fig. 1 General Arrangement of Conventional Double Wall L.N.G. Tank.

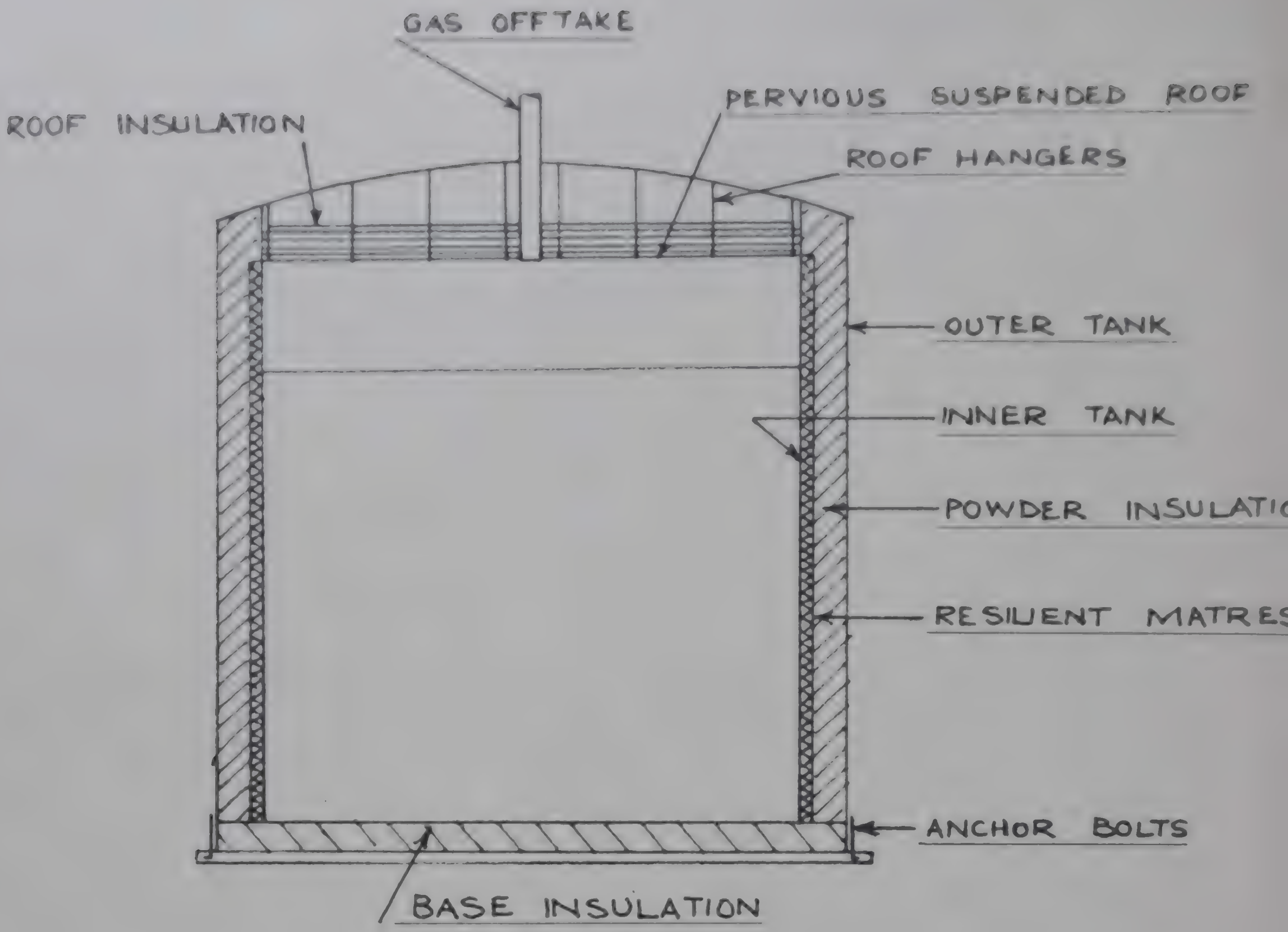

GAS OFFTAKE
PERVIOUS SUSPENDED ROOF
ROOF INSULATION
ROOF HANGERS
OUTER TANK
INNER TANK
POWDER INSULATI
RESILIENT MATRES
ANCHOR BOLTS
BASE INSULATION

Fig. 2 General Arrangement of L.N.G. Tank with Suspended Roof.

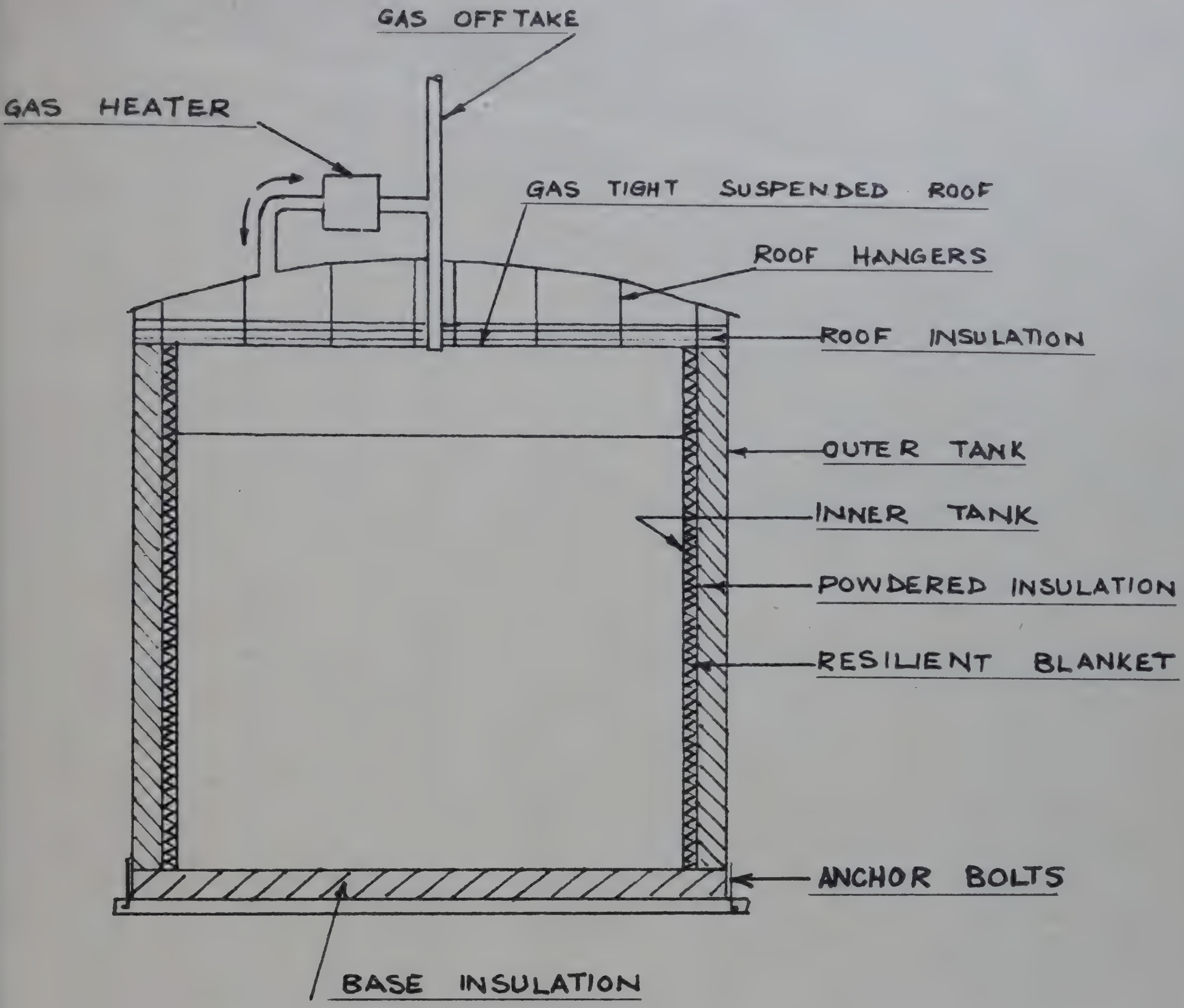

GAS OFFTAKE
GAS HEATER
GAS TIGHT SUSPENDED ROOF
ROOF HANGERS
ROOF INSULATION
OUTER TANK
INNER TANK
POWDERED INSULATION
RESILIENT BLANKET
ANCHOR BOLTS
BASE INSULATION

Fig. 3 General Arrangement of L.N.G. Tank with externally Pressurised Suspended Roof.

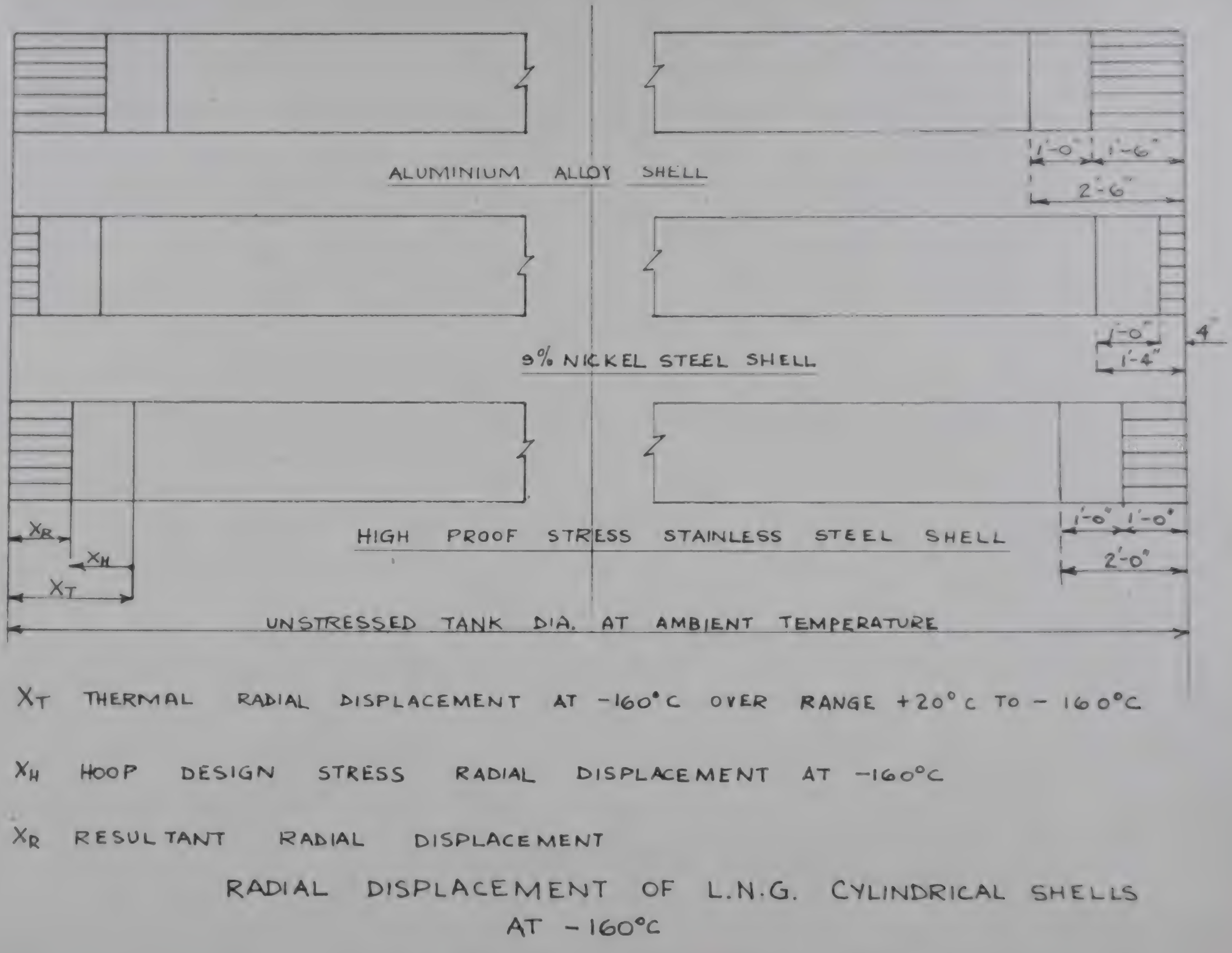

1'-0"
1'-6"
2'-6"
ALUMINIUM ALLOY SHELL
1'-0"
4"
1'-4"
9% NICKEL STEEL SHELL
X_R
X_H
X_T
HIGH PROOF STRESS STAINLESS STEEL SHELL
1'-0"
1'-0"
2'-0"
UNSTRESSED TANK DIA. AT AMBIENT TEMPERATURE
X_T THERMAL RADIAL DISPLACEMENT AT -160°C OVER RANGE +20°C TO -160°C
X_H HOOP DESIGN STRESS RADIAL DISPLACEMENT AT -160°C
X_R RESULTANT RADIAL DISPLACEMENT
RADIAL DISPLACEMENT OF L.N.G. CYLINDRICAL SHELLS
AT -160°C

Fig. 4 Typical Radial displacement of L.N.G.

DESIGN CONSIDERATIONS FOR LNG STORAGE TANKS

CURRENT UNITED STATES PRACTICE

By R. W. Bodley *

Introduction

Above ground, metal, double wall tanks are the most practical means for storing liquefied natural gas today. This is because they provide both structural reliability and predictability of boil-off rates at acceptable costs. Significant differences between the design of LNG storage tanks and other low temperature tanks have required the extension of existing technology. This paper discusses some of the important design considerations. It gives current approaches to and solutions of design problems in the United States. An attempt is made to present pertinent technical information on the major storage components. The paper places emphasis on those items not readily found in current literature and by-passes those which have already been well covered.

Standards

The main reference standard in the United States covering details of design for LNG tankage is a recently issued appendix to the American Petroleum Institute Standard 620, "Design and Construction of Large, Welded, Low Pressure Storage Tanks". This standard is Appendix Q, "Low Pressure Storage Tanks for Liquefied Natural Gas", and is referred to often in this paper.

Another reference which covers the storage and handling of LNG is Standard No. 59A-T of the National Fire Protection Association entitled "Standard for the Storage and Handling of Liquefied Natural Gas at Utility Gas Plants". However, it does not give detailed design and erection requirements, but rather references API Standard 620. Still another reference is the recently issued API Standard No. 2510A, "Design and Construction of LNG Installations at Petroleum Terminals, Natural Gas Processing Plants, Refineries, and Other Industrial Plants". Although these standards cover certain aspects of storage design in some detail, there is no combination of standards that gives all the information required for a safe, dependable storage system. Companies designing and erecting LNG storage tanks rely on a combination of research and testing backed by experience and good judgment. Examples of design considerations not covered by standards will be found later in this paper.

API 620, Appendix Q

Appendix Q of Standard 620 allows both single and double wall construction,

* Graver Tank & Mfg.C.

but this discussion is limited to double wall because it is the only type being built today. In double wall cryogenic storage, the insulating space is maintained at a positive pressure to insure a reliable vapor barrier. This positive pressure is maintained in two ways: One design uses a nitrogen purge gas while the second, a more recent approach, uses boil-off LNG. With a nitrogen purge, the vapor space of the inner tank is sealed from the annulus while with a boil-off gas purge, it need not be.

To set design criteria and approaches for the inner tank, the components of LNG storage tanks have been categorized in Appendix Q as either Primary or Secondary. Primary components are defined as those which may be stressed to a significant level, the failure of which would permit leakage of the liquid being stored, those exposed to a refrigerated temperature between -60F and -270F, and those subject to thermal shock. Secondary components, on the other hand, are those which will not be stressed to a significant level by the refrigerated liquid, the failure of which will not result in leakage of the liquid being stored, and those exposed to product vapor having a design metal temperature of -60F or higher. Figure 1 depicts a standard double wall design with a nitrogen purge and shows the components considered to be Primary and Secondary.

Material requirements then, as well as many other design considerations, are based upon this designation, Primary or Secondary, and the type of annulus purge. Major inner tank components, roofs, shells, bottoms, stiffeners, and fittings are limited to 9% Nickel, aluminum, or stainless steel. Major outer tank components can be carbon steel, usually A283, Grade C in the United States when nitrogen purging is used. However, when boil-off gas purging is used, the outer tank materials must be selected on the basis of design temperature. This may be governed either by minimum atmospheric temperature or by the boil-off gas temperature, depending upon the features of the particular design. Also, for boil-off gas purging, outer tank welds must conform to the lap or butt welded construction of Standard 620. In terms of current practice, this essentially means that the outer shell must be full fusion, double butt welded, and spot X-rayed as contrasted to single butt welding, outside only or double butt welding without full fusion and with no X-ray when a nitrogen purge is used.

Bottom Insulation

Properly designed and installed insulation systems are as important to the success of LNG storage as the tank components themselves; however, these systems are not covered by codes or standards. Insulations placed between the inner and outer tank bottoms must have the combination of structural strength to carry the loads imposed by the inner tank and its contents and good insulating value.

This combination is difficult to obtain since the first requires high density material while the second demands low density. It is common practice to use a ring of insulating material with good load bearing capability under the inner tank shell to transmit the concentrated loads of the inner tank to the foundation. Inside this ring, where the loads are not so concentrated, a material with a better insulating value, but with lower, yet adequate capability to carry the imposed loads, is used.

One system of bottom insulation is shown in Figure 2. The primary insulating material, covering say 90% of the bottom, is foam glass block. A layer of sand is placed first on the outer bottom to provide a level working surface. Then the layered block and another sand layer are placed. The upper sand layer protects the block from damage while the inner bottom is being layed. The sand should be screened before placement and should be chemically inert so it will not cause corrosion. Depending upon its source and moisture content, it may need to be dried before placing. Sheets of asbestos or felt paper are placed between the layers of foam glass block to limit deflection on initial loading. If this paper or some other filler material is not used, the surface of the block breaks down causing a slight settling until an adequate bearing area is reached.

To transmit the more concentrated loading of the inner shell, a ring of perlite concrete or pre-cast perlite concrete blocks is used. This is usually a 27 lb. per cu. ft. density mixture. Special attention should be paid to heat transfer in this area so that outer tank temperatures remain within design limits. It is essential to the success of a bottom insulation system that it be kept dry during construction. The best method to accomplish this is to complete the outer tank before placing the bottom insulation. There is a considerable amount of research and testing being presently conducted on insulating materials, and designers will have an expanding choice in the future.

Sidewall and Roof Insulation

Tank sidewalls and roofs are usually insulated with expanded perlite because of its low installed cost and good insulating properties. The fact that perlite can be expanded at the tank site or expanded nearby and trucked to the site, then blown into the annulus is a significant factor in its low in-place cost. It is also significant from the standpoint of the control of the moisture content and thereby, the K factor. It is installed hot and dry soon after expansion and therefore does not pick up moisture from the atmosphere.

There is a considerable amount of literature available on expanded perlite insulation for use at low temperatures. Thermal conductivity data are readily available for various grades of perlite. The Perlite Institute, Inc., has published technical data sheets on a variety of topics on perlite. Their Technical Data Sheet No. I-3 entitled "Specifications, Expanded Perlite Insulation for Low Temperatures in Atmospheric Service" gives the physical properties that must be specified and controlled to obtain a quality insulation system. These are: grading, density, moisture content, and organic material content. This specification also covers methods of testing and sampling from bagged perlite. For field expanded perlite, tank manufacturers have developed their own specifications in cooperation with their perliters, and sampling and inspection are carefully detailed. Insulating a one-billion standard cubic feet LNG tank may take as long as four weeks of continuous filling. During this time, hourly samples are usually taken and checked for conformance to specifications. Of course, the annular space must be clean and dry before insulating begins and the annulus pressure maintenance system must bee activated soon after the insulating is completed.

Perlite Pressure on Warm-Up

An important problem to the tank designer is the loading imposed on both the inner and outer tank shells by the perlite insulation. The most critical loading occurs on the assumed condition of taking the tank out of service and warming it back up to atmospheric temperature after it has been in service for some time. Perlite insulation will compact and settle to some extent from thermal cycling of both the inner and outer shells. On warm-up as the inner shell expands, the perlite compresses, putting an internal pressure on the outer shell and an external pressure on the inner shell. The inner shell does not return to its as-built position, but rather the outer shell expands some, the perlite compresses and an intermediate position is reached. The pressure developed at this position is dependent upon a number of factors, including the temperature change, the thermal coefficient of expansion of the inner shell, the inner and outer tank dimensions which establish the width of the annulus, the height of the insulation, and the properties of the perlite.

Therefore, since both the inner and the outer shells must be designed for the in-service and warm-up conditions, a thorough knowledge of the behavior of perlite under these conditions is necessary. Like all materials, perlite deflects under load, and like other granular materials, its behavior can be evaluated by its physical and mechanical properties. These properties are the angle of repose, angle of internal friction, angle of friction on metals, density, and modulus of elasticity. Data on several of these properties are not found in the literature at all, and the published data are not related to the in-service or warm-up condition.

Graver has combined field observations with laboratory determination of the properties of perlite to develop this data and relate it to storage conditions. The important facts established from these studies are:

1. Perlite will compact and settle in service to some extent from thermal cycling of both the inner and outer shells and any vibrations present.

2. This normal settlement is not harmful, provided:

   a. There is sufficient reservoir of perlite above to make up for the settlement so that bare spots will not occur and topping off will not be required.

   b. The increase in K factor caused by increased density is considered in the design.

3. The maximum settlement that can occur in service with even the most severe vibration present is approximately 20 per cent.

4. A survey of existing installations shows that the maximum settlement that has occurred in the United States on refrigerated and cryogenic tankage is about 10 per cent.

5. The maximum pressure can be predicted and should be used in the design to give an insulation system of infinite life regardless of the number of thermal cycles.

Once the governing design pressure is determined, it is then a matter of economics as to the best method to provide for it. We have found the most economical choices to be stiffening the inner tank shell or increasing the width of the annulus or a combination of the two.

Self Supporting Roofs

LNG storage tanks, even of very large diameters, are built with self supporting roofs primarily because column loadings on the tank bottom would create bottom restraints, resulting in high thermal stresses during cooldown. Design and erection technology of these free span roofs have been the subject of considerable research, testing and improvement by individual tank companies in the United States. Graver has been designing and building free span roofs and domes for a number of years. Some of our early experience was with geodesic and modified lamella domes designed as industrial buildings. In studying structural arrangements for these domes, it was apparent that most designs involved a focal point at the center with structural members radiating in some pattern from it. Also, from these studies, the realization that a grid pattern would reduce the amount of structural required emerged. The grid pattern is not new, but it is simple. Our studies also disclosed that if the grid members were layered, the strength was not decreased and the intersections would be much simpler and cheaper to construct. By rolling or pressing the grid arches to great circle arcs, and layering the top arches over the bottom arches, construction is simplified and strength can be developed by bracing the arches to one another, rather than framing them into one another on a single plane. This two-layer grid concept was further developed by model analyses and actual construction and testing of tank roofs. Extending this technology and experience with tank and industrial roofs into the fields of refrigerated and cryogenic storage was a natural step. Figure 3 shows a 194 ft. diameter outer roof of an LNG storage tank being constructed.

To further simplify erection problems and reduce costs, these roofs are built on supports on the tank bottom and air lifted into position when the tank shell is completed. Our first experience with a large air lift was in 1959 on a 384 ft. dome weighing over 700 tons. Air lifting large tank roofs requires first a design to provide a positive sliding seal to the tank shell, then a mechanism to stabilize the roof and keep it level on its ascent, and finally a means to secure the roof safely in position when it reaches the top of the tank. Blowers mounted in shell or roof manholes provide air at the required rate and pressure.

Air lifting large free span tank roofs has been quite successful and economical. Graver has lifted many, the largest being the 194 ft. diameter roof shown under construction in Figure 3. Great care has been taken in preparation and controlling each lift and because of this each lift has been accomplished without incident. The ability to design and construct such free span roofs helps make the storage of LNG in double wall, metal tanks possible and practical.

Quality Control and Testing

A lot of care is taken in controlling quality of all tank components as fabrication and erection proceed. Then when the tank is complete, each component is subjected to one or more strength and leak checks. These requirements are spelled out in detail in API Standard 620 and its Appendix Q. Weld examination in primary components is accomplished by both X-ray and die penetrant examination. Vertical shell seams can be either spot or full radiographed and the design joint efficiency factor is determined by the amount of radiography, 85% for spot and 100% for full examination. The most common practice, however, is to fully X-ray the vertical seams. The savings in material and welding costs are normally more than enough to justify the additional X-ray. Horizontal seams can be spot X-rayed, but some users ask for full X-ray of all shell seams. Die penetrant examination is used to check welds in primary components that are not X-rayed. This includes examination of fitting welds, welds of stiffeners, compression girders, anchor bolt chairs, and the like.

Impact testing of all 9% Nickel steel is required, as well as impact testing of the welding procedure and the welder and operator qualification. Material qualification impacts are Charpy V-notch, transverse. All impact tests are run at -320°F. This temperature was selected so that liquid nitrogen can be used for cooling the impact specimens.

API 620 Appendix Q requires that material for primary components be marked to relate each component to its mill test report. This requires effective control of in-process work. In addition on 9% Nickel steel tanks, since it is not readily distinguished from carbon steel by eye, Graver requires a final test of each piece of the primary components to make sure material was not misplaced.

Strength and leak test procedures for the completed tank are carefully conducted and include the following major steps:

1. Bottom seams of both the inner and outer tanks are leak tested by vacuum box.

2. Inner tank shell to bottom fillet welds are leak tested by air pressure and soap suds.

3. Outer tank shell to bottom fillet welds are tested by either air pressure and soap suds or by penetrating oil.

4. The inner tank is filled with water to the equivalent pressure of the full head of LNG. Appendix Q allows this head to be increased by 25% if considered desirable, but limits tank shell stresses to a maximum of 80% of the specified minimum yield strength of the material, or 50% of the specified minimum tensile strength.

5. The inner tank is then pressurized in several stages to a maximum of 1¼ times the design pressure.

6. The pressure is then reduced to the design pressure and shell and

roof seams above the liquid level are tested with soap suds. Opening pressure of the pressure vent is checked at this time.

7. While the inner tank is being emptied, the opening pressure of the vacuum vent is checked.

8. With the tank empty, the inner tank is again pressurized to design pressure and the anchor system is rechecked.

9. The outer tank is then pressurized to its design pressure and is tested for leaks with soap suds. As an alternate to this, the outer tank shell seams can be checked by penetrating oil. Also, outer tank roof seams can be checked by vacuum box.

10. Operation of outer tank pressure vents is checked during this test, and the vacuum vent is checked by applying the design vacuum.

Conclusion

The American Petroleum Institute Standard 620, Appendix Q is the main reference for the design and construction of LNG storage tanks in the United States. It covers many important design considerations and requirements for dependable storage of LNG in above ground, metal tanks. However, several of these considerations are not covered in detail and tank manufacturers have relied on their own research and testing to provide needed information. A considerable amount of research is still being conducted to advance the technology even further.

References

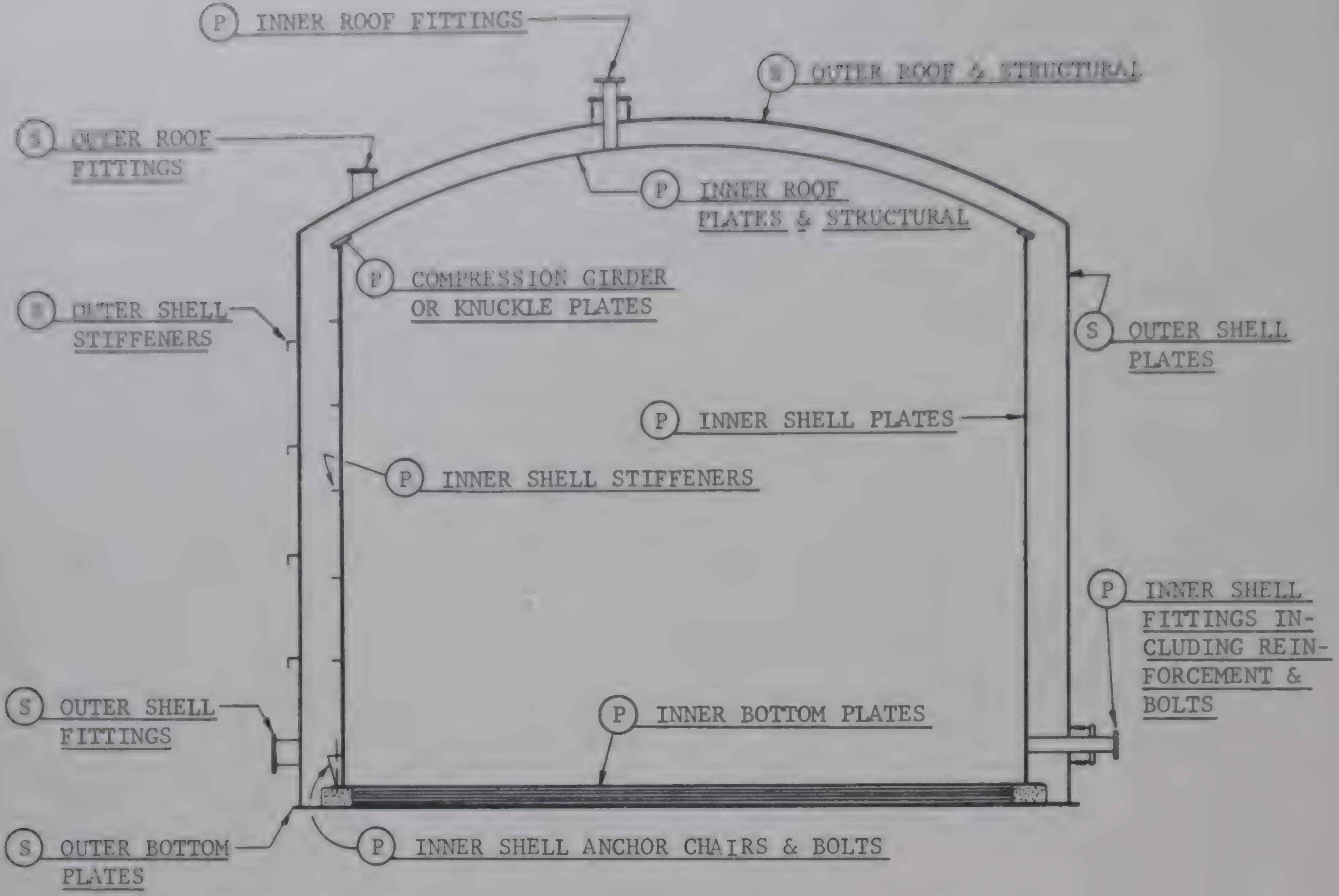
1. API Standard No. 620 ---- Design and Construction of Large, Welded Low-Pressure Storage Tanks

   Appendix Q ---- Low-Pressure Storage Tanks for Liquefied Natural Gas

2. NFPA No. 59A-T-1966 ---- Standard for the Storage and Handling of Liquefied Natural Gas at Utility Gas Plants

3. API Standard No. 2510A ---- Design and Construction of LNG Installations at Petroleum Terminals, Natural Gas Processing Plants, Refineries, and Other Industrial Plants

4. Perlite Institute, Inc. Technical Data Sheet No. 2-3 ---- Expanded Perlite Insulation for Low Temperatures in Atmospheric Service


P INNER ROOF FITTINGS
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S OUTER ROOF FITTINGS
P INNER ROOF PLATES & STRUCTURAL
P COMPRESSION GIRDER OR KNUCKLE PLATES
S OUTER SHELL STIFFENERS
S OUTER SHELL PLATES
P INNER SHELL PLATES
P INNER SHELL STIFFENERS
P INNER SHELL FITTINGS INCLUDING REINFORCEMENT & BOLTS
S OUTER SHELL FITTINGS
P INNER BOTTOM PLATES
S OUTER BOTTOM PLATES
P INNER SHELL ANCHOR CHAIRS & BOLTS

Ⓟ PRIMARY COMPONENTS - 1) STRESSED TO SIGNIFICANT LEVEL
2) FAILURE WOULD PERMIT LEAKAGE
3) EXPOSED TO REFRIGERATED TEMPERATURE
4) SUBJECT TO THERMAL SHOCK

Ⓢ SECONDARY COMPONENTS - 1) NOT STRESSED TO SIGNIFICANT LEVEL
2) FAILURE WOULD NOT PERMIT LEAKAGE
3) EXPOSED TO PRODUCT VAPORS AND HAVING DESIGN TEMPERATURE OF -60 F OR HIGHER

Fig.1. Double wall L.N.G.tank - nitrogen purge primary and secondary components

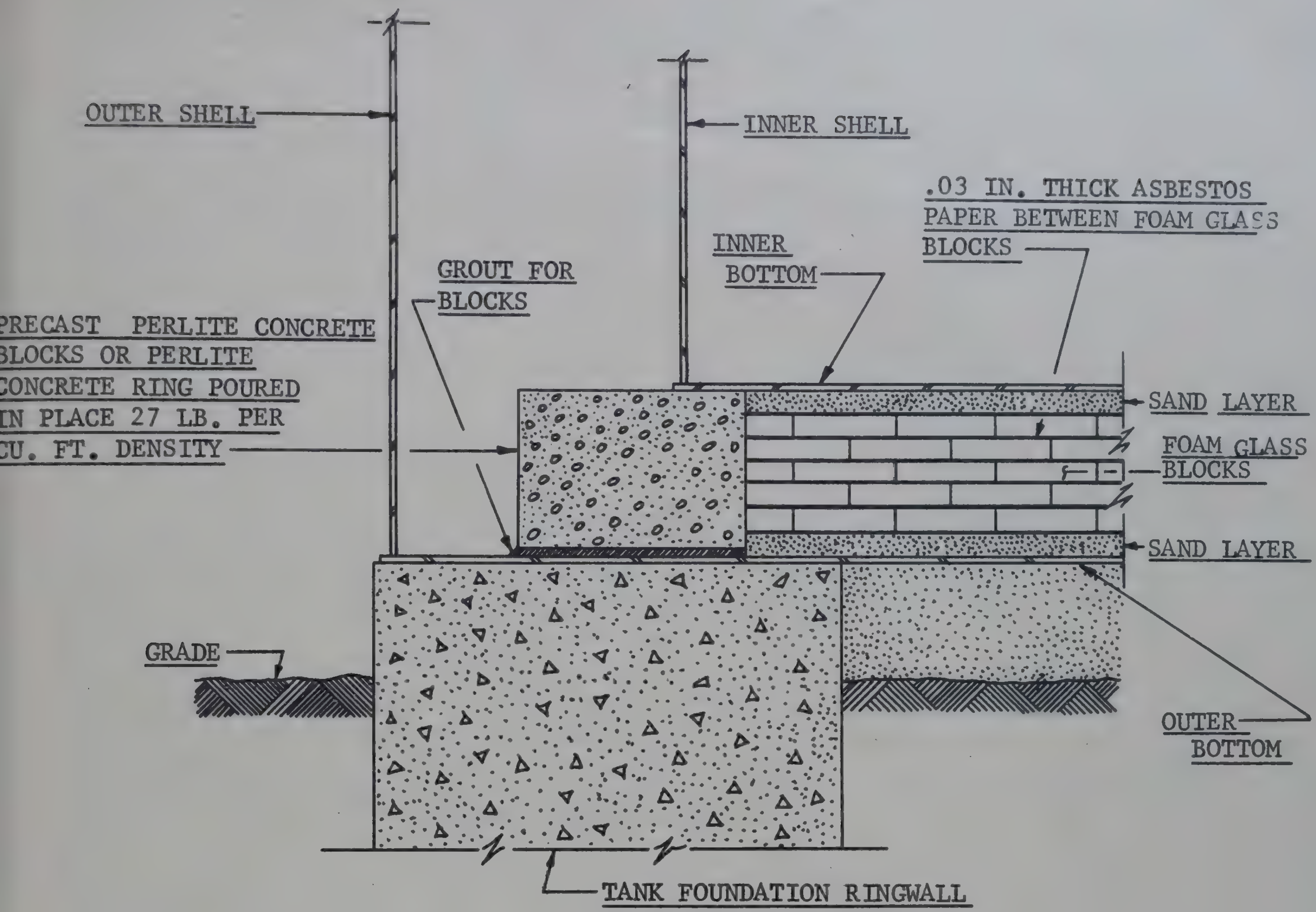
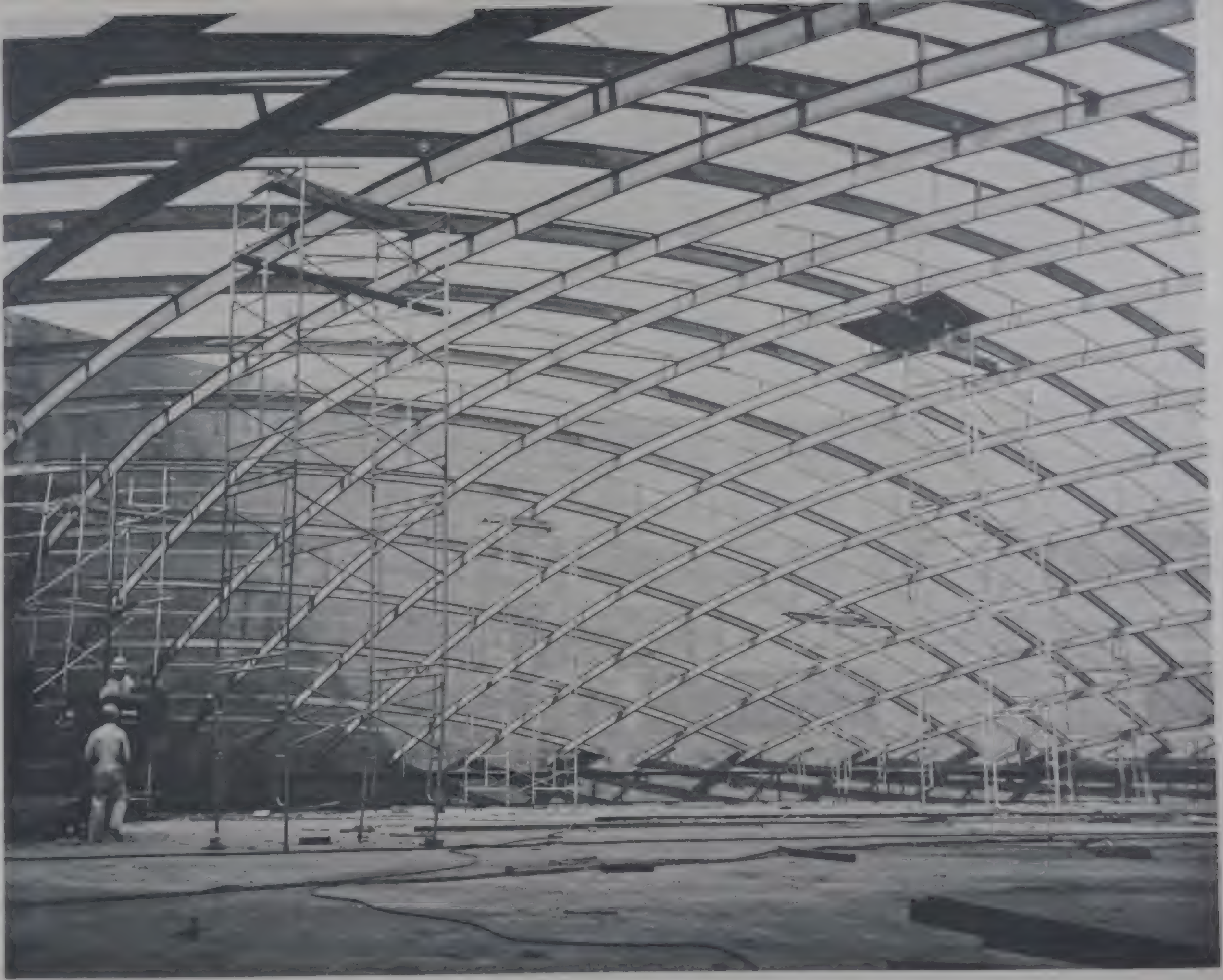

OUTER SHELL
INNER SHELL
.03 IN. THICK ASBESTOS PAPER BETWEEN FOAM GLASS BLOCKS
INNER BOTTOM
GROUT FOR BLOCKS
RECAST PERLITE CONCRETE LOCKS OR PERLITE ONCRETE RING POURED N PLACE 27 LB. PER U. FT. DENSITY
SAND LAYER
FOAM GLASS BLOCKS
SAND LAYER
GRADE
OUTER BOTTOM
TANK FOUNDATION RINGWALL

Fig.2. Insulation system for tank bottom



EVALUATION OF THE FATIGUE STRENGTH OF INTERGRATED TANKS FOR LNG SHIPS

By D. J. Burns*, R. G. Jackson+ and J. G. Kalbfleisch¤

SUMMARY

The trend in design of ships for transporting liquefied natural gas is towards the replacement of self-supporting tanks by integrated tanks. The latter are thin-metallic membrane tanks supported by load-bearing insulation fitted to the inside of the hull.

This paper describes one design of integrated tank, discusses the loads imposed on it and shows how the result of cyclic pressure fatigue tests on pilot scale tanks have been analysed statistically using the likelihood function. The paper also shows how the fatigue data can be combined with the sea load spectrum to predict whether ship tanks will have an acceptable fatigue strength.

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INTRODUCTION

Liquefied natural gas (LNG) is transported by sea in specially designed double-hull ships. It is carried at atmospheric pressure in tanks separated from the inner side of the inner hull by insulation material. The cargo tanks of these ships are of two types:

Self-supporting - in which the tank is designed to withstand, without recourse to the ship structure, all of the static and dynamic loads produced by the cargo.

Integrated - in which the tank is a thin metallic membrane supported by load-bearing insulation fitted to the inside of the inner hull.

This paper describes one design of integrated tank, discusses the loads imposed on it and shows how the results of fatigue tests on pilot scale tanks have been analysed statistically using the likelihood function. The relative merits of the likelihood function compared to more conventional methods of statistical analysis are discussed.

DESCRIPTION OF INTEGRATED TANKS

The integrated tank referred to in this paper is an improved version of the membrane used in the first membrane LNG tanker, Pythagore (1) combined with a modified version of the insulation system used in the first self-supporting LNG tankers, Methane Princess and Methane Progress (2). Fig. 1 shows a section of a right-angle corner of this design.

A number of layers of balsa, glued together, form the core of the insulation. This core is faced with plywood to make insulation panels which are bonded to hardwood grounds, fixed to the double hull in a rectangular pattern. The spaces between the grounds are packed with mineral wool. Insulation continuity is ensured between sandwich panels by means of expanded polyvinyl-chloride wedges, compressed and bonded to the panels. These wedges are covered with plywood. The plywood assembly is covered by a facing of balsa which is machined to give a smooth, flat surface onto which is fitted the membrane. This whole assembly acts as a secondary barrier for the LNG if there should be a serious leak in the membrane.

The thin stainless-steel membrane is an assembly of corrugated sheets. The sheets, which are lap-welded together, have wave-shaped corrugations in two perpendicular directions. One wave is slightly smaller than the other and at their intersections they form a complex expansion joint known as a knot. Each set of corrugations is parallel to one of the ship's axes and therefore to one of the tank axes. Along the edges of the tank (i.e. along the junction of any two walls) the corrugations on adjacent walls are joined by angle pieces with a corrugation which includes a special corner knot, Fig. 1.

Along the edges of a tank the balsa facing pads are replaced by hardwood blocks glued onto the plywood facing of the insulation panels. Rigid stainless-steel angle pieces are bolted to these hardwood blocks. The edges of the membrane sheets are welded to these rigid angle-pieces so that when the tank is cooled the edge loads are transferred through the insulation to the ship structure.

STATIC AND CYCLIC LOADS CARRIED BY THE MEMBRANE

When a tank is cooled, thermal stresses are induced in the membrane because of the restraint at the edges and the thermal gradient across the load-bearing insulation.

At sea the wave motion produces two types of cyclic mechanical stress in the tanks. Cargo motion produces a cyclic pressure spectrum which can be related to histograms showing the probability of occurence of various sea states and the resulting bow accelerations of the ship (3). These cyclic pressures which apply a combination of cyclic bending and pressure to the knots have the most serious effect on the knots near the edges of the tank and which straddle the junction between the stainless-steel angle and hardwood facing the pads. Hull bending produces a cyclic elongation and contraction spectrum in the membrane which can be estimated by a method similar to that used for the pressure spectra (3).

FATIGUE TESTS

An extensive fatigue test programme has been carried out on this membrane design to assess the effect of thermal, pressure and hull bending strains acting separately or together (3, 4). An important part of this test programme was cyclic pressure tests on pilot scale tanks made from full-size components of membrane and insulation using the same installation procedures as on board ship. Such tests show the effects of irregularities of construction and necessary tolerances on the fatigue strength of the membrane and provide a useful cross check with fatigue data obtained from component testing (3, 4).

It was impracticable to carry out low temperature fatigue tests on such large tanks (5 m^3) because the highest practical speed is low (200 c.p.m.) and the requirement to test to lives of several million cycles therefore involves long test programmes. Thus the membrane was pretensioned to simulate both the mean strain imposed by cooling down and a still-water bending moment.

It was decided to carry out each test at a constant amplitude and obtain data for two pressure amplitudes. These amplitudes were 16 ± 9 psi and 16 ± 12 psi. These correspond respectively to the mean of the highest one tenth and the mean of the highest one hundredth accelerations experienced by a 40,000 m^3 ship in service. For a 20 year ship life there will probably be no more than 82,000 cycles of pressure greater than 16 ± 9 psi and 1000 cycles of pressure greater than 16 ± 12 psi.

Fig. 2 shows the exterior of one of the test boxes. The pumps are mounted on the roof which is removable. The membrane has to be modified in the vicinity of the roof to provide a seal; thus failures along the top edges are not representative of the situation in a ship tank. These "non-significant" failures make it difficult to determine the precise moment during a test when a "significant" failure occurs in the membrane so that the test must be stopped periodically while all the knots are examined for failure and repairs are made.

ANALYSIS OF FATIGUE DATA

Fig. 3 is a projection of the membrane surface of a test box. There have been no significant failures in knots lying over balsa pads or in the corner knots. The tests have shown that the areas most susceptible to failure are the knots straddling the junctions between the stainless steel angles and hardwood facing pads. This is perhaps not surprising since there is less yield in the supporting structure at these points and any inaccuracy of assembly is more likely to produce stress concentrations. It was found that failures were likely to occur at two points on these knots, Fig. 4; either at the lower point of intersection of the corrugations (type γ) or in the fold joining this point to the top of the knot (type β). The former predominantly occur over the stainless steel and the latter over the hardwood.

A test box contains 44 knots which straddle the junction between the stainless steel angles and hardwood facing pads. A second failure in a knot is discounted since covering the first failure with a patch alters the stiffness of the knot. Thus a test box yields m – 1 stopping times of c_1, c_2, ..., c_{m-1} (millions of cycles) and f_i is the number of failures observed to occur between c_{i-1} and $c_i (1 \leq i \leq m - 1)$. If f_m is the number of knots which have not failed at the conclusion of the test then $f_1 + f_2 + \ldots + f_m = 44$. Table 1 gives the data for type failures in a box with m – 1 = 7 stops and $f_8 = 32$ unfailed knots after 4×10^6 cycles at a pressure amplitude of 16 ± 9 psi.

It is very difficult to analyse by conventional statistical methods (5,6) such grouped data from a severely truncated distribution. However, the likelihood function (7,8) provides a straightforward way of handling this data and its use is illustrated here.

i	1	2	3	4	5	6	7	8
c_i	1.810	2.344	2.588	2.743	3.272	3.582	4.000	∞
f_i	0	4	5	1	0	2	0	32

Table 1

It is necessary to assume a shape for the fatigue life distribution of the knots. Two distributions commonly used in fatigue analyses are the Weibull and Log-normal (5,6) but neither of these can account for the cluster of failures between 1.81 and 2.59×10^6 cycles and the large number of unbroken knots at the completion of testing. An approach which works well for this and other data is to assume that for the endurance range considered (up to 4×10^6 cycles) the distribution of failures is a constant p times a log-normal distribution, where $0 \leq p \leq 1$.

Swanson (9) has proposed a model which gives a reasonable explanation for the success of this approach. A sketch of the pressure amplitude-fatigue life (S-N) curve corresponding to his model is shown in Fig. 5. It is postulated that there are two separate failure distributions, "early" failures and "late" failures. Let p be the proportion of early failures in a population. At high pressure amplitudes $p \rightarrow 1$ and at low pressure amplitudes $p \rightarrow 0$. Swanson found that at intermediate pressure amplitudes there was a bimodal distribution with well separated modes and he postulated that both modes followed a log-normal distribution.

In the present study the quantity of primary interest is T_α, the lifetime exceeded with probability $1-\alpha$ for small α and moderate to large pressure amplitudes. Clearly only the early mode of a bimodal distribution is of practical interest and the present tests were stopped before the late mode appeared. Consequently, it has not been necessary to assume a shape for the "late" distribution; it could be log-normal or a very long tail.

Let p_i be the probability that a particular knot fails between c_{i-1} and c_i, where $c_o = 0$. Then under our model:

$$P_i = p \left\{ \Phi\left(\frac{\log c_i - \mu}{\sigma}\right) - \Phi\left(\frac{\log c_{i-1} - \mu}{\sigma}\right) \right\} , \quad 1 \leq i \leq m-1 \qquad (1)$$

$$P_m = 1 - p_1 - p_2 - \cdots - p_{m-1}$$

where μ and σ^2 are the mean and variance of the log-normal distribution, and $\Phi(t)$ is the normal integral

$$\Phi(t) = \int_{-\infty}^{t} \frac{e^{-u^2/2}}{\sqrt{2\pi}} \, du \qquad (2)$$

Let T_α be the lifetime exceeded with probability $1 - \alpha$, where $\alpha < p$. Then:

$$\frac{\log T_\alpha - \mu}{\sigma} = Z_\alpha(p) \qquad (3)$$

where, for given α and p, Z is obtained by solving the equation:

$$p \, \Phi \, (Z_\alpha \, (p) \,) = \alpha \qquad (4)$$

Solving Eq. (3) for μ and substituting into Eq. (1), we express p_i in terms of the three unknown parameters T_α, σ^2, p:

$$p_i = p \left\{ \Phi \left(\frac{\log [c_i/T_\alpha]}{\sigma} + Z_\alpha(p) \right) - \Phi\left(\frac{\log(c_{i-1}/T_\alpha)}{\sigma} + Z_\alpha(p) \right) \right\}, \quad 1 \leq i \leq m-1 \qquad (5)$$

$$p_m = 1 - p_1 - p_2 - \cdots - p_m$$

Now assume that the chance of failure at one knot is unaltered by the failures of other knots. Then the probability of the observed frequencies f_1, f_2, ..., f_m, is:

$$L(T_\alpha, \sigma, p) = \frac{(f_1 + f_2 + \dots + f_m)!}{f_1!\, f_2! \dots f_m!} \; p_1^{f_1} \, p_2^{f_2} \dots p_m^{f_m} \qquad (6)$$

This probability depends upon the unknown parameters T_α, σ, and p, and is called the likelihood function for T_α, σ, and p on the basis of the observed frequencies f_1, f_2, ..., f_m. It is denoted by $L(T_\alpha, \sigma, p)$.

For any specified parameter value (T_α, σ, p), Eq. (6) gives the probability of the observations made. Values of (T_α, σ, p) for which Eq. (6) is large will be preferred to values for which Eq. (6) is small, because the former values make the observed frequencies more probable. Thus the likelihood function Eq. (6) may be used to rank various parameter values (T_α, σ, p) in order of their plausibilities.

The parametric value $(\hat{T}_\alpha, \hat{\sigma}, \hat{p})$ for which Eq. (6) is a maximum is called the maximum likelihood estimate of (T_α, σ, p). At this parameter value, the observed frequencies have the maximum probability possible under the model. Any other parameter value (T_α, σ, p) may be compared with $(\hat{T}_\alpha, \hat{\sigma}, \hat{p})$ by means of the relative likelihood function.

$$R(T_\alpha, \sigma, p) = \frac{L(T_\alpha, \sigma, p)}{L(\hat{T}_\alpha, \hat{\sigma}, \hat{p})}, \quad 0 \quad R \quad 1$$

If R is near 1, the value (T_α, σ, p) is quite reasonable in that it makes the observed frequencies almost as probable as the model permits. However, if $R(T_\alpha, \sigma, p)$ is small, the value (T_α, σ, p) is implausible because there exist parametric values near (T_α, σ, p) which make the observed frequencies much more probable. Thus by examining the relative likelihood function one may determine which parametric values are comparatively plausible and which implausible.

Note that the above statements apply to the three parameters T_α, σ, p, simultaneously. Here T_α is of primary interest, while σ and p are of secondary importance. In order to determine whether a particular value $T_\alpha = T_\alpha^{\,o}$ is implausible, one may find the maximum over all σ and p of $R(T_\alpha^{\,o}, \sigma, p)$. If the maximum is small, the set $(T_\alpha^{\,o}, \sigma, p)$ is implausible for every σ and p, and thus $T_\alpha^{\,o}$ is not a plausible value for T_α. On the other hand, if the maximum is near 1 there exist values of σ and p for which the set $(T_\alpha^{\,o}, \sigma, p)$ is fairly likely, and thus $T_\alpha^{\,o}$ cannot be ruled out.

Thus the plausibilities of various T_α values may be obtained by plotting or tabulating the maximum of R over σ and p as a function of T_α. It is usually more convenient to work with the natural logarithm of R,

$$r(T_\alpha) = \underset{\sigma,\ p}{\text{maximum}} \log R(T_\alpha, \sigma, p)$$

As an example take $\alpha = .01$ and consider the data of Table 1. Table 2 gives the maximum log relative likelihood $r(T_\alpha)$ for several values of T_α, together with the values of σ and p for which the maximum is attained. Figure 6 is a plot of $r(T_\alpha)$ against T_α. For each T_α, the maximum of log R was found using a modified Newton-Raphson method on an IBM360-75 computer. The computations required less than 30 seconds of computer time in all, and more efficient programming could reduce this time substantially.

From Table 2 and Figure 6 it is concluded that values of $T_{.01}$ between 1.41 and 2.16 (million cycles) are all reasonable in the light of the data of Table 1. For such values the maximum relative likelihood is at least 0.1, so that for some (σ, p) values the observed frequencies are at least one tenth as probable as they can be under the model. Values of $T_{.01}$ less than 1.08 or greater than 2.26 (million cycles) are quite implausible in that for no values of σ and p are the observed frequencies more than one one-hundredth as probable as they can be under the model. Similar computations may be carried out to shed light on the range of plausible values of σ and p.

$T_{.01}$	$r(T_{.01})$	σ at max	p at max
1.0	-5.18	.497	.367
1.1	-4.49	.426	.339
1.2	-3.76	.370	.323
1.3	-3.02	.323	.311
1.4	-2.27	.284	.303
1.5	-1.55	.251	.297
1.6	-0.91	.223	.291
1.7	-0.39	.198	.285
1.8	-0.06	.176	.278
1.9	-0.03	.157	.270
2.0	-0.43	.140	.258
2.1	-1.43	.127	.243
2.2	-3.28	.117	.222
2.3	-6.20	.113	.192

Table 2

Computations for the data of Table 1

It has been claimed that a serious drawback to the use of likelihood as a measure of uncertainty is that it is not probability and so is not intuitively obvious to some people. It has, however, a definite interpretation in terms of probabilities and grouped or truncated data are handled easily within the framework of the theory.

A likelihood is a probability statement about the observations actually made in an experiment. The information provided by two independent experiments is easily combined by multiplying likelihood functions from the individual experiments to give overall likelihood. In contrast a confidence coefficient is a probability statement about the observations to be expected in the long run if the experiment is repeated over and over. For further discussions and examples of drawing inferences from the likelihood function the reader is referred to (7,8).

PREDICTION OF THE FATIGUE LIFE OF SHIP TANKS

As in the testing of large sections of aircraft structures (10, 11, 12) it is not possible to determine a complete endurance curve because of the large number of specimens required to make probability statements. It is accepted practice to determine the endurances at one or two load levels for a specified T_a, e.g. A and B on Fig. 5, and to assume from experience or component testing a general shape for the endurance curve e.g. ABC on Fig. 5.

This endurance curve and the cyclic pressure spectrum corresponding to the required ship life can then be used with the Palmgren-Miner Cumulative Damage theory (13) to assess whether the ship tanks will have an acceptable fatigue strength. This theory assumes that the proportion of fatigue life used up when n_i cycles of a particular cyclic plus static load are applied, may be expressed by $n_i/T_{\alpha i}$ where $T_{\alpha i}$ is the number of cycles likely to be exceeded (with probability $1 - \alpha$) before fatigue cracking occurs. When a number of load levels are applied the life used up is:

$$\frac{n_1}{T_{\alpha 1}} + \frac{n_2}{T_{\alpha 2}} + \cdots \qquad \text{or} \quad \Sigma \frac{n}{T_\alpha}$$

and failure is assumed to occur when:

$$\Sigma \frac{n}{T_\alpha} = 1$$

Tests on structural components ranging from simple notched bars to aircraft wings (14) have shown that the fatigue lives of specimens subjected to spectrum loading are usually longer than those predicted from the normal type of S-N fatigue curve (each specimen tested at constant load amplitude) using the Palmgren-Miner rule. Factors of two or three are not uncommon but in a few cases the fatigue life has been almost half the predicted value. Unfortunately, none of the more sophisticated cumulative damage theories (13, 15) have been shown to be more consistent in the accuracy with which they predict fatigue life and the Palmgren-Miner rule is still widely used in design studies in the United Kingdom and elsewhere.

The application of an analysis of this type to the results of cyclic pressure tests on the pilot scale tanks suggest that $\Sigma \frac{n}{T_\alpha} \leqslant 0.12$ for a 40,000 m^3 ship, which corresponds to a fatigue life far in excess of the normally accepted working life of 20 years. This and other studies (3, 4) have been submitted to Bureau Veritas, Lloyd's Register of Shipping and United States Coast Guard, who have given their approval in principle to the use of this type of integrated tank in LNG tankers.

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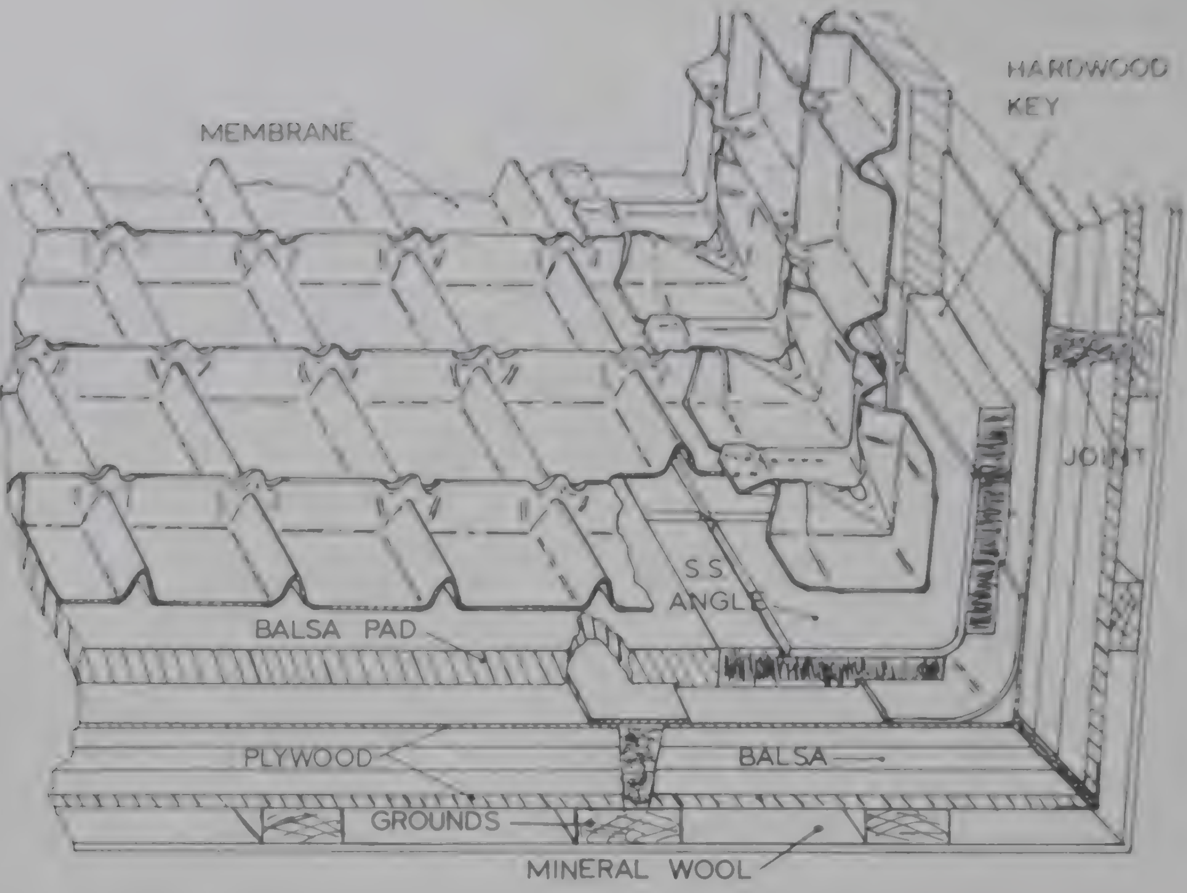
MEMBRANE
HARDWOOD KEY
JOINT
S.S. ANGLE
BALSA PAD
PLYWOOD
BALSA
GROUNDS
MINERAL WOOL

Fig.1

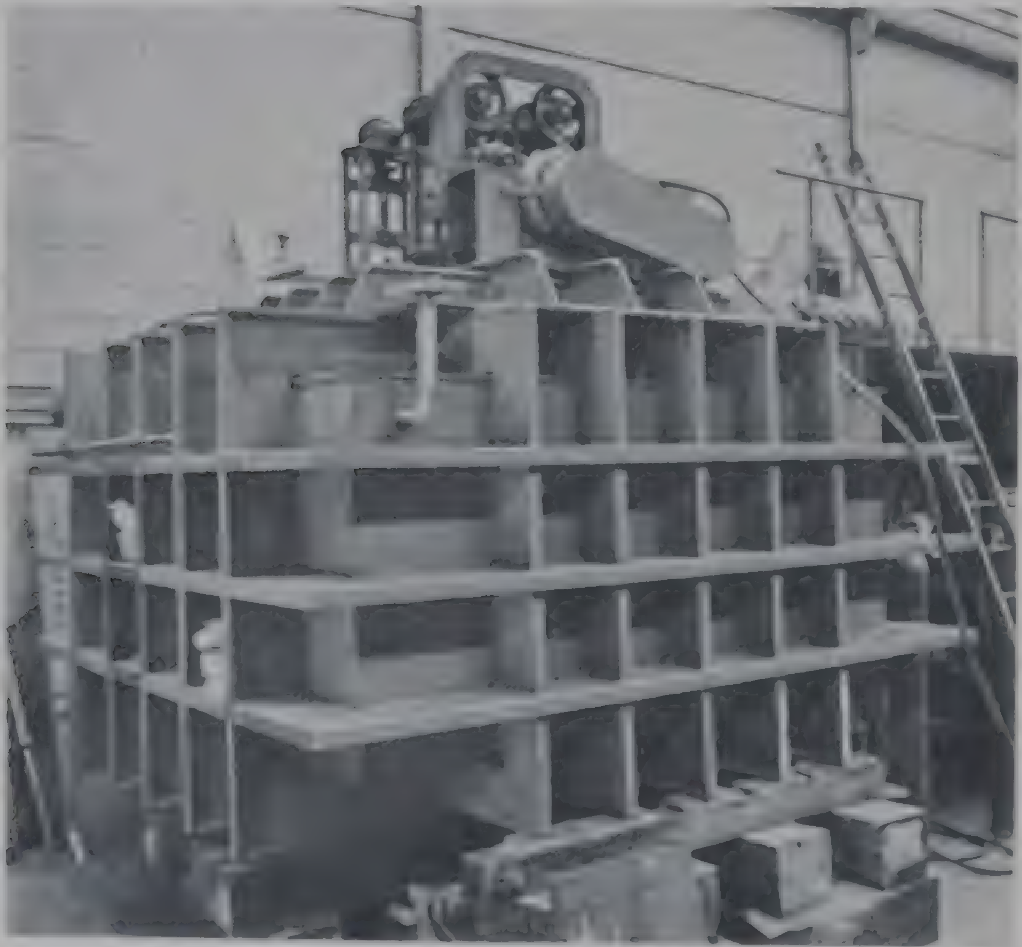

Fig.2

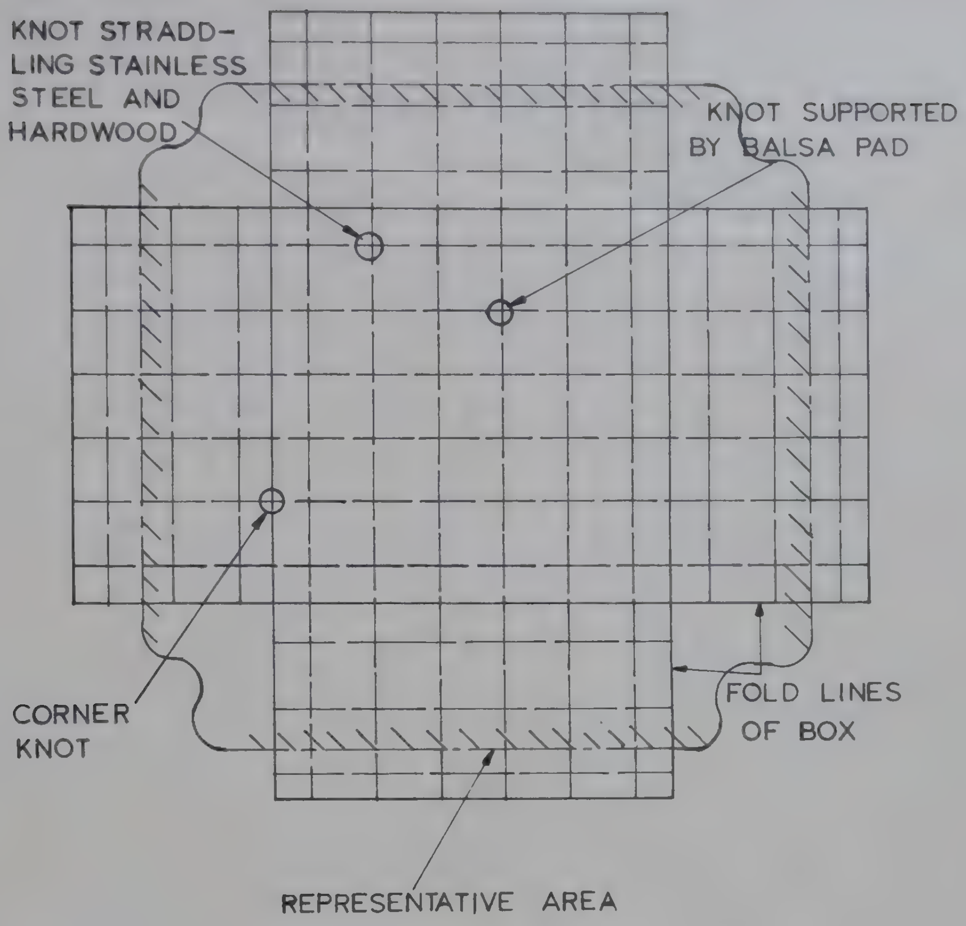
KNOT STRADDLING STAINLESS STEEL AND HARDWOOD
KNOT SUPPORTED BY BALSA PAD
CORNER KNOT
FOLD LINES OF BOX
REPRESENTATIVE AREA

Fig.3

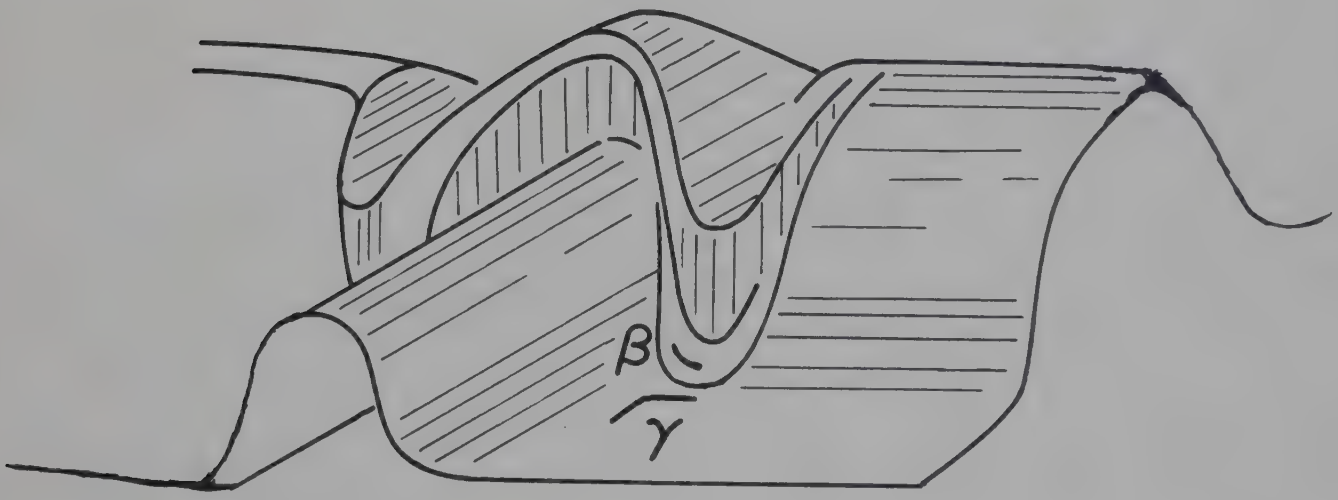
β
γ

Fig.4

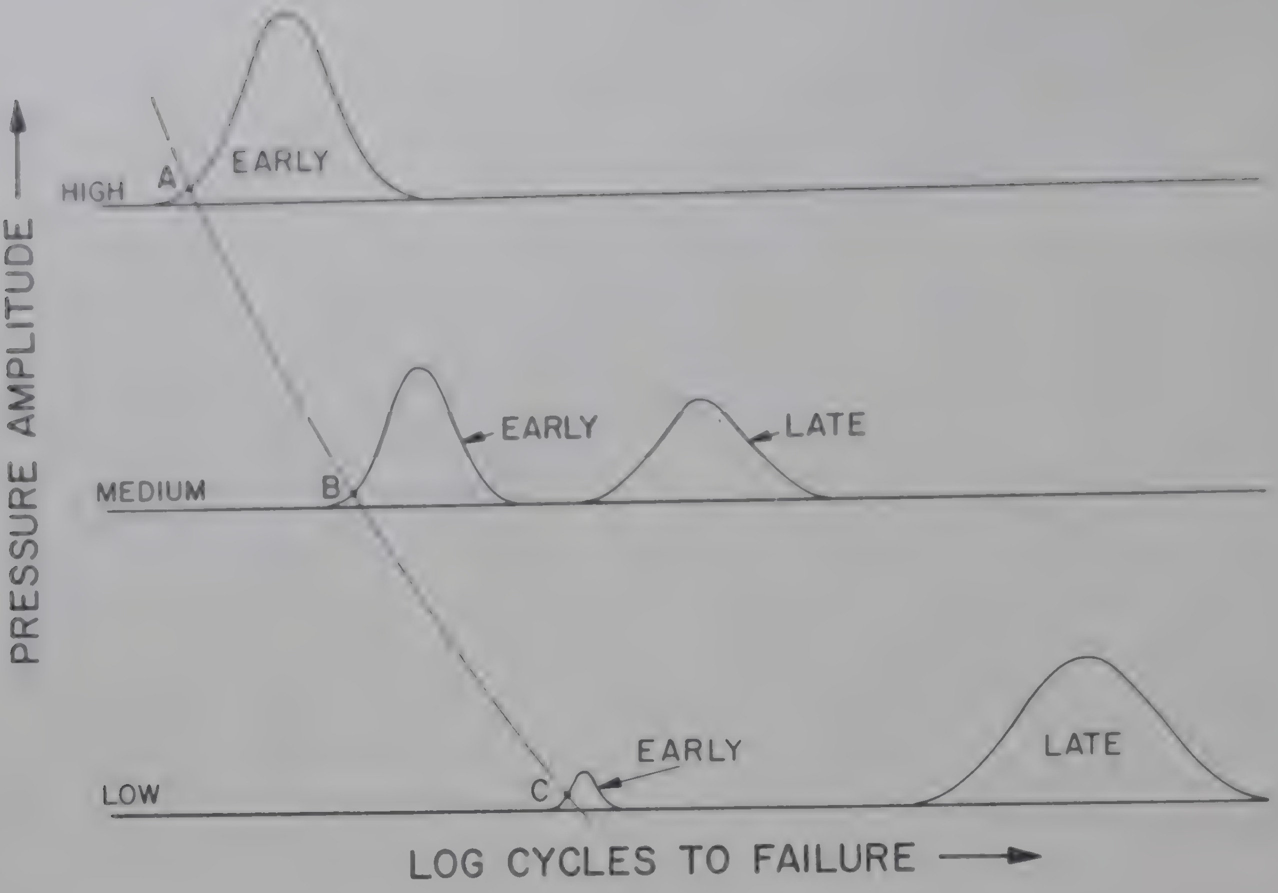
PRESSURE AMPLITUDE
HIGH
A
EARLY
MEDIUM
B
EARLY
LATE
LOW
C
EARLY
LATE
LOG CYCLES TO FAILURE

Fig.5

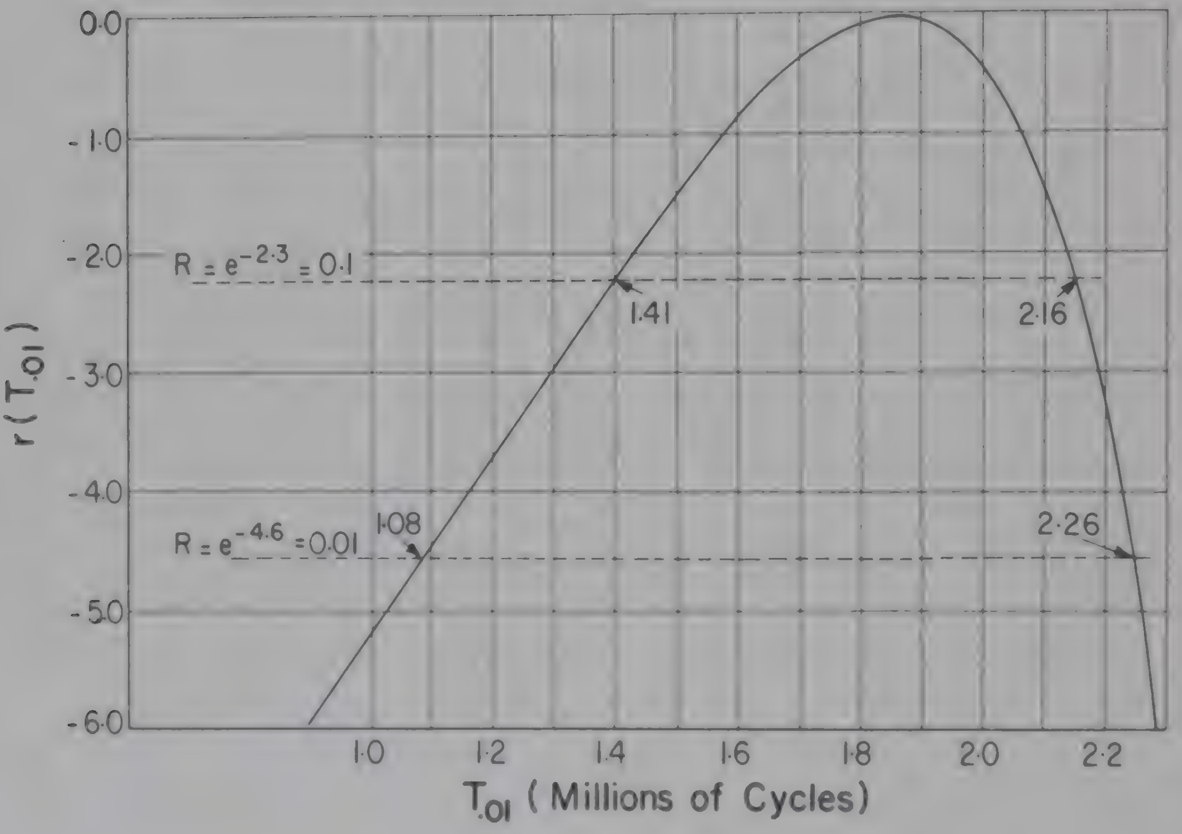
0.0
-1.0
-2.0
-3.0
-4.0
-5.0
-6.0
r(T.01)
R = e^-2.3 = 0.1
1.41
2.16
R = e^-4.6 = 0.01
1.08
2.26
1.0
1.2
1.4
1.6
1.8
2.0
2.2
T.01 (Millions of Cycles)

Fig.6

"A New Economic Process for LNG Shipping-Liquefaction and Regasification Cold Use"

V.C.Williams,St. Louis,Mo.,O.H.Simonds,New York, N.Y.,W.M.Williams, Pittsburgh, Pennsylvania

Summary

Natural gas liquefaction and transportation from a field site to a market site is accomplished by transporting liquid nitrogen or liquid air by ship from the market place to the field site where it is used to liquefy the natural gas. The availability of the cold or refrigeration in the liquid nitrogen or air reduces the equipment required at the field site. Liquid natural gas is pumped into the holds of the same cryogenic tanker ship used to transport the liquid nitrogen or air and the liquefied natural gas is returned to the market site. Regasification of the liquid natural gas is carried out to use the cold or refrigerant effect therein to liquefy nitrogen or air, which is then charged to the returning tanker to the field site. The process is a repetition of liquid nitrogen or liquid air from the market site to the field site and there used to liquefy natural gas, which is then shipped from the field to the market where the liquid natural gas is gasified and in the process liquefies nitrogen or air for return to the field. The cycle employs about 1.0 to 2.1 pounds of nitrogen to 1.05 pounds of methane. In the process of both the field site and the market site, various cycles employing expansion of the various high pressure streams for cooling are employed and the work is matched against compression of other streams and subsequent expan-

sion for efficiency in the system and minimization of power requirement.

For a 390,000,000 (STP) cu ft/day gas transportation project, the estimated cost for liquefaction and regasification (exclusive of gas cost, shipping, and profit) is $0.0649/Mcf, compared to an estimate for the conventional cascade cycle of $0.2060/Mcf.

Introduction

There is a challenge to harness for economic use, the abundance of unused natural gas in areas until now beyond the reach of the world's markets. For in addition to unassociated gas produced from gas wells, vast quantities of natural gas are automatically produced in association with crude oil. Much of this production occurs in the developing countries where, because of distance or intervening seas, it cannot be transported economically by pipeline to the world's industrial complexes. Because of this, only a fraction of the world's natural gas resources are at present usefully consumed. The vast bulk is left to lie idle in the gasfields for lack of possibility of developing large scale local uses on an economic basis, or in the case of gas associated with oilwells, it is flared.

It is one thing to release natural gas at great pressure, it is another to reduce it to manageable proportions, transport it, store it, and sell it in competition with other fuels.

The best way to deal with a large volume of gas is to liquefy it. However, natural gas, which is largely methane, cannot be liquefied by simply increasing the pressure, as has been the case with heavier hydrocarbons used for energy purposes. The critical temperature of methane is -116.5°F. This means that methane can only be liquefied below that temperature regardless of the pressure applied. At atmospheric pressure, the liquefied gas must be kept at or below its boiling point of -258°F. Hence, the techniques of liquefying and handling natural gas have been intimately related to advances in cryogenic technology.

Basically, the liquefaction of natural gas requires the removal of energy in the form of sensible and latent heat. This process can be accomplished by:

1) Stage mechanical refrigeration where heat is transferred through refrigerants to a high level heat sink. This method uses what is referred to as a cascade cycle or process.
2) Compressing and expanding the gas, using turbo-expanders. This is known as an expander cycle.

The most widely used cycle is the cascaded vapor compression system. This cycle, based on refrigeration principles, uses commercially available refrigerants whose thermodynamic and physical properties are fairly well-known.

Two areas of interests have indicated a definite need for large-volume storage of LNG:

1) In transporting gas from energy-rich to energy-poor areas in insulated tankers, large storage depots at both the loading and the market port are required. In such base-load operations, the storage facility is essentially a surge tank smoothing out the nonuniform base-load

demand and fuel delivery operations. In this case, it is not necessary to preserve the LNG in the tank for long periods, since it is passed on to distribution within a short period of time.

2) The continued growth and expansion of the natural gas industry has resulted in the need for storage of large volumes of gas near metropolitan areas to meet winter peak-loads. In this case, LNG is stored for relatively long periods of time and used during only a few days of the winter.

A great deal of research is at present being devoted to finding and refining industrial processes which will make continuous use of the very large amounts of cold available. The main possibilities here lie in the development of major industrial applications such as the production of liquid oxygen, nitrogen, hydrogen, etc.

Ideally, to make the maximum use of cold potential, it is necessary that such use should show a considerable reduction in capital investment compared with conventional processes for making some saleable commodity.

The cold should replace low-temperature refrigeration which is very expensive compared to mild refrigeration (e.g. the low temperatures needed in the production of cryogenic liquids). The overall cost of refrigeration in the liquefaction of the natural gas is a major part of the final sales price.

There is, however, another manner of approaching the use of the available "cold". This would be to consider the "cold" as an inherent part of the process thus making it the commodity; not selling it or something derived from it, but building its use into the process technology to reduce the investment and operating costs and thus to increase the profitability of a natural gas distribution venture. Williams-Simonds have devised a workable method to accomplish this. Liquid nitrogen or air as an intermediary is transported by ship from a market site to the remote field site and there is used to liquefy the natural gas. The availability of this cold reduces greatly the equipment in the field. The liquid natural gas (LNG) is then pumped into the holds of the same cryogenic tanker ship which brought the cargo of liquid nitrogen or air and the LNG is returned to the market site. For regasification of the LNG the "cold" is used to liquefy nitrogen or air which then becomes the return cargo for the tanker. The process then becomes a constant repetition of: liquid nitrogen or air from market to field to use to liquefy natural gas which is shipped field to market where the LNG is gasified and liquefies nitrogen or air to return to the field, etc., etc., etc.

The process seems best applicable to the base-load type of operation. With a major portion of the total investment in ships and land base storage, travel and turn-around time are significant factors in the economic use of these transport items. Multiple re-use of these facilities leads logically to the high throughput consideration which is the "flat" characteristic of base loading. The process has patents allowed or patents pending in a total of 35 nations.

Discussion

Many reviews of liquefied natural gas technology have been published (References 3, 4, 10). To reduce a gas at ambient temperature (or above its critical temperature) and at some pressure to a liquid at some low temperature at atmospheric pressure requires in a continuous flow process either or both:

a) the removal of heat from the gas system.

b) the performance of net work by the gas system.

In the notation of Keenan (Ref. 6) the availability of the gas in steady reversible flow must decrease with reference to ambient to reduce the gas to a liquid. In the more familiar thermodynamic property and process description (Ref. 2,7), for a flow process and in the absence of effects of change in kinetic energy, potential energy, electrical and magnetic fields, surface effects, etc.,

$$\Delta H = q - w_{shaft}$$

ΔH = change in enthalpy of the system in flow (outlet - inlet)

q = heat input

w_{shaft} = net work output

For liquefaction of the gas system, ΔH must be negative. For a no-work output system q must be negative which is the same as a) above. Similarly, for an adiabatic work producing process it is b) and there may be combinations of these. In the Linde-Hampson simple Joule-Thomson air liquefier with neither heat nor work flow the input air stream is compressed to a high pressure to have a lower enthalpy than the outlet countercurrent exchanger air at the same temperature. This works, of course for air, but not, e.g., for hydrogen at ambient temperature.

The classic in the development of heat removal or a) type process was the cascade liquefier of Pictet (Ref. 9) with the changes made by Onnes (Ref. 8). The process is used in many LNG installations, as, for example, in the plant at Arzew, Algeria (Ref. 3).

The "work producer" or expansion processes of type b) required the growth of a mechanical technology before their application. Claude (Ref. 1) developed a successful reciprocating compressor. Many trials at the problem of a centrifugal expander had partial success, but with Kapitza's work (Ref. 5) on the centripetal radial inflow machine and stability maintenance the turbo-expander began its new life and most large plant use turbo-machines to develop work output thus cooling the gas. The work output is usefully employed to drive a necessary compressor or auxiliary work absorber.

For a continuous acting liquefaction plant it follows from the second law of thermodynamics as noted by Obert (Ref. 7) and Dodge (Ref. 2) that the greatest net work output and, ipso facto: the greatest liquefaction, when both compressors and turbo-expanders are in the flow system will occur when:

1) expansions through the turbines take place from as high a useful temperature as possible and,
2) compression is performed from as low a temperature as is justified.

One of us (Ref. 11) employed these precepts in a method for gas liquefaction and rectification.

The conceptual processes that we have developed employ these concepts in a novel manner. The patents (Ref. 12) which have issued are covered by broad claims. Fundamentally the cycles use turbo-expanders which drive rotary compressors of the axial flow or centrifugal type. They are of a new generation of high rotative speed machines with isentropic efficiencies in the 80-90% range (Ref. 10). Direct connected from driver to driven, and without gear trains, parasitic losses are minimized. The machines are stable, small-sized, and have a low cost per horsepower.

Process Description

The specific gravity of LNG or liquid methane is about one-half that of liquid air or nitrogen (the refrigerant). If the ship transport is first considered, the tank volume would permit the lading of twice the mass of refrigerant compared to the mass of LNG. This is the extreme for any cycle design (2.1 lb LNG per lb refrigerant) but it is not a reasonable or proper ship design, for the ship would have about twice the cargo tonnage, full, outbound from the field as inbound to the field full of refrigerant. The hydrostatic head on the tanks is also different requiring some excesses for the LNG tank design. These criteria indicated that a shipping ratio of LNG to refrigerant near one is desired.

The question may be raised as to the return shipment of refrigerant in tanks half-full or half the tanks full and how this affects ship stability. Our ship consultants advise that the preference would be for tanks half-full. Longitudinal swash bulkheads would reduce the free surface effect in the tanks to a negligible quantity. Stability in terms of metacentric height would be maintained. The ships being considered have an LNG capacity of 385,000 barrels, API (26,000 metric tons) and are estimated to cost $21,500,000 each.

These ship design factors, thus, generally set the LNG to refrigerant mass ratio at about one and the cycles were devised on this ratio. Evaporation losses will occur in transport and storage but these may be used for fuel or recovered in a Stirling engine cycle. Nevertheless, losses have been included in the cycle summaries at five per cent.

Field Site

The field site process is shown in Figures 1 and 2. Natural gas is arbitrarily assumed to be available at 800 psia. As one method of start-up, the compressor-turbine combinations are put on natural gas operation, by using by-pass valves, until a storage charge of LNG has been accumulated. During regular operations, any make-up refrigerant flow may be taken from the effluent and by a small auxiliary compressor put into the inlet of compressor C-1. It is of some importance to note that this cycle is operable on air as the refrigerant which thus obviates the shipment of refrigerants to the field site as required in the cascade cycle.

The plant is of a relatively small size and may be built as a shipboard installation.

Market Site

The market cycle is shown in Figures 3 and 4. Just as in the field site process, the thermal loads in the heat exchangers are adjusted so that real temperature differences exist to obey the second law of thermodynamics. This cycle also may be used for a start-up procedure using the refrigerant compressor (CN) and expander (TN) or an auxiliary to accumulate liquid refrigerant.

General Comments

The refrigerant may be either nitrogen or air. To obtain nitrogen, the market cycle would be operated with air but this air would be rectified within the cycle to separate the oxygen as a reject stream. For every 1.0 lb of LNG processed, 0.308 lb of 95% oxygen would be produced. The quantity of oxygen realized is sufficient to support a satellite process industry in the market site.

If, however, liquid air is the transferred refrigerant purging of the vessel will be required in the market site. The vapor methane in the tank would be purged with nitrogen from rectified air. This purging would require rectification of 2.5% of the amount of liquid air to be shipped to obtain vapor nitrogen.

In the field site after complete pump-out of the liquid air from the ship tanks, the vapor would be 95% nitrogen. Methane is non-flammable with this gas. Purging of the vessel under these conditions at the field site does not appear to be necessary.

Economics

Plant cost estimates were made based on manufacturers' quotations. From these and using best judgment for labor, depreciation, rate-of-return on investment, shipping charges, and miscellaneous processing costs the following table shows the estimated price structure for gas service to the distribution point:

Basis

Field gas purchases:

389,000,000 cfd

(142,000,000 Mcf/yr)

Market gas sales (after deducting fuel and losses):

363,000,000 cfd

(133,000,000 Mcf/yr)

Summary

Cost of service in table as $/Mcf

Gas Cost @ Field/Mcf

Distance, n-miles (ships req'd)	$0.05	$0.10	$0.15
2000 (4)	0.30	0.35	0.40
4000 (7)	0.39	0.45	0.50
6000 (10)	0.49	0.54	0.60

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    U.S. Pat. 2,713,781 (July 26, 1955)
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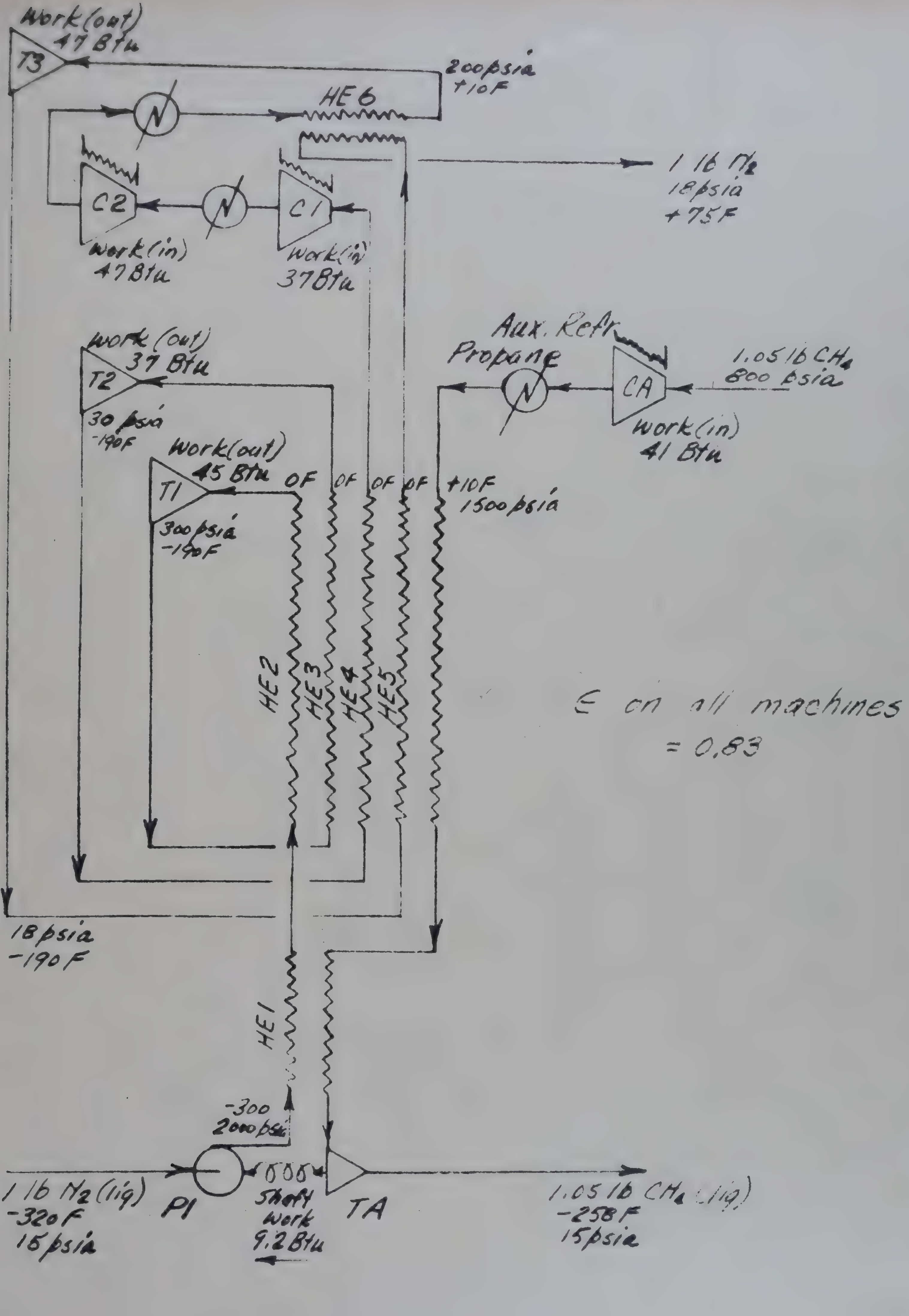
Work(out) 47 Btu
T3
200 psia +10F
HE 6
C2
C1
work(in) 47 Btu
work(in) 37 Btu
1 lb N_2 18 psia +75F
work (out) 37 Btu
T2
30 psia -190F
Aux. Refr. Propane
CA
1.05 lb CH_4 800 psia
work (in) 41 Btu
work(out) 45 Btu
T1
300 psia -190F
0F
0F
0F
0F
+10F 1500 psia
HE2
HE3
HE4
HE5
ϵ on all machines = 0.83
18 psia -190F
HE1
-300 2000 psia
1 lb N_2 (liq) -320F 15 psia
P1
Shaft work 9.2 Btu
TA
1.05 lb CH_4 (liq) -258F 15 psia

Fig.1. Field site process

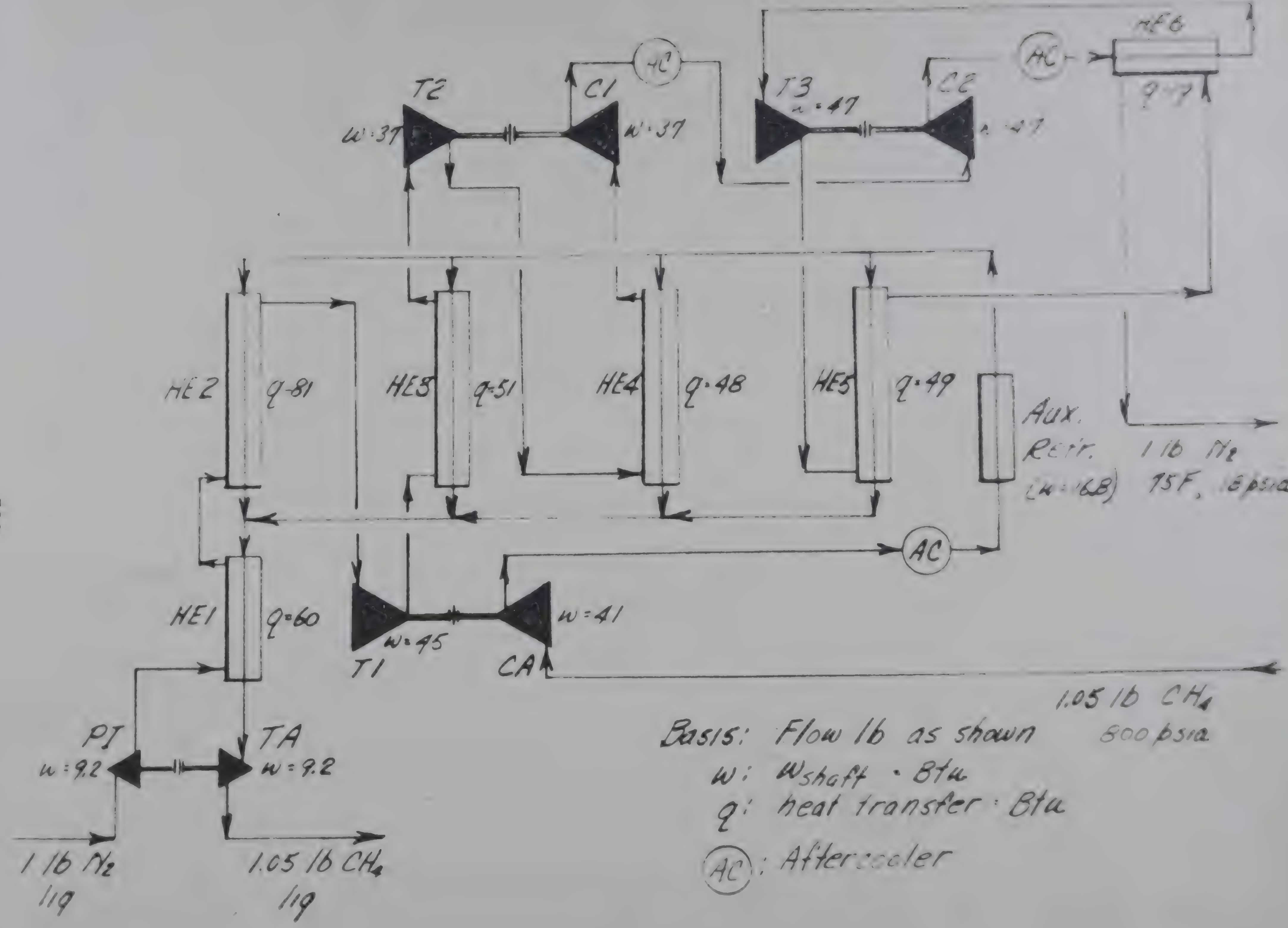
T2
w=37
C1
w=37
AC
T3
w=47
C2
w=47
AC
HE6
q=7
HE2
q=81
HE3
q=51
HE4
q=48
HE5
q=49
Aux.
Refr.
(w=16.8)
1 lb N_2
75F, 18psia
HE1
q=60
T1
w=45
CA
w=41
AC
1.05 lb CH_4
800 psia
PT
w=9.2
TA
w=9.2
1 lb N_2
liq
1.05 lb CH_4
liq
Basis: Flow lb as shown
w: W_{shaft} · Btu
q: heat transfer · Btu
AC: Aftercooler

Fig.2. Field site process block diagram

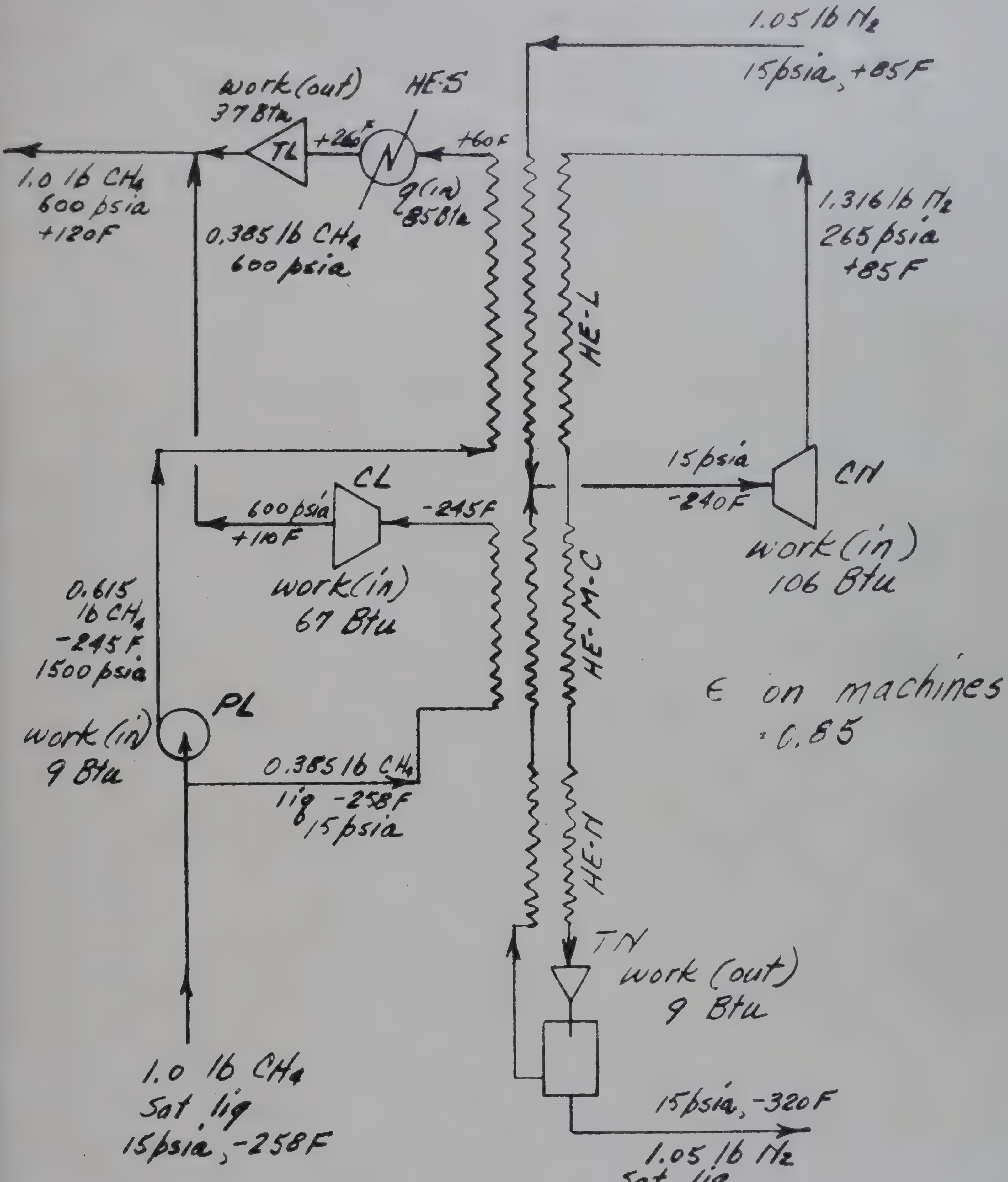
1.05 lb N_2
15psia, +85F
work (out)
37 Btu
HE-S
TL
+260F
+60F
q (in)
85 Btu
1.0 lb CH_4
600 psia
+120F
0.385 lb CH_4
600 psia
1.316 lb N_2
265 psia
+85F
HE-L
15 psia
-240F
CN
work (in)
106 Btu
CL
600 psia
+110F
-245F
work (in)
67 Btu
0.615
lb CH_4
-245F
1500 psia
HE-M-C
ϵ on machines
= 0.85
PL
work (in)
9 Btu
0.385 lb CH_4
liq -258F
15 psia
HE-N
TN
work (out)
9 Btu
1.0 lb CH_4
Sat liq
15psia, -258F
15psia, -320F
1.05 lb N_2
Sat liq

Fig.3. Market site process

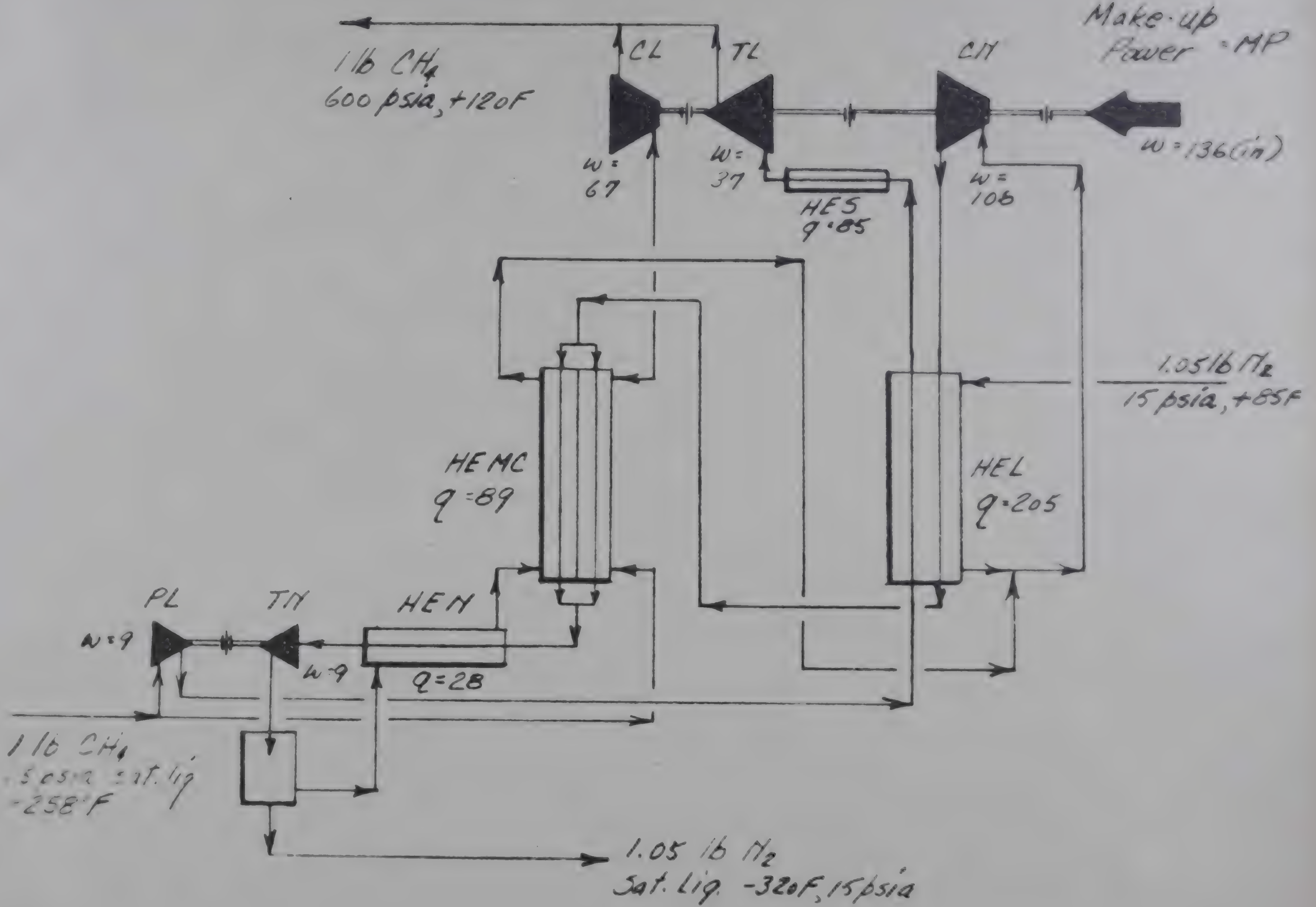
Make-up Power = MP
w = 136 (in)
CL
TL
CM
1 lb CH4
600 psia, +120F
w = 67
w = 37
w = 108
HES
q = 85
HEMC
q = 89
HEL
q = 205
1.05 lb N2
15 psia, +85F
PL
TM
HEM
w = 9
w = 9
Q = 28
1 lb CH4
15 psia sat. liq
-258°F
1.05 lb N2
Sat. Liq. -320F, 15 psia

Fig.4. Market site process block diagram

LIQUEFIED NATURAL GAS PIPELINES FOR ARCTIC GAS RECOVERY

By G.Walker*, D. Coulter*,and N.Sood*

ABSTRACT

The present search for oil in the Canadian North may reveal the existence of large gas fields and a market for at least 2 b.c.f.d. can be foreseen in Eastern Canada and the United States. The gas may be pipelined South for transmission through existing trunk-lines or to a Northern railhead for shipment in the liquid state using insulated tank cars.

This paper is concerned with three pipeline systems for bringing the gas South namely

(a) conventional natural gas transmission lines

(b) long distance liquid natural gas pipelines

(c) evaporative two phase non-isothermal pipelines

No definitive conclusions are presented as studies are at an early stage.

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INTRODUCTION

In the Department of Mechanical Engineering of the University of Calgary a broad programme of fundamental research on liquefied natural gas is in progress. One part of this programme is concerned with developing the technology of liquefied natural gas (LNG) pipelines both for short distance transfer type operation and for long distance transportation.

Elsewhere Norrie and Walker (1) have discussed LNG pipelines in general and the Calgary programme in particular. Here this paper is concerned with a potential application of LNG pipelines to transport of gas from the Canadian North.

Northern Gas

Consideration of the established oil and gas deposits continuing North from the Gulf of Mexico through the United States into Alberta leads to the expectation that similar deposits exist in Northern Canada including the Yukon Territory, the North West Territories and the Arctic Islands. The various oil and gas shows in these areas, the established operations at Norman Wells, N.W.T.,and the recent major oil strike on the North Slope of Alaska all support this expectation. The high level of interest in these deposits is evident from the substantial explorations now in progress by a consortium of oil and gas companies supported in part by the Canadian Government.

Although the principal interest of these explorations is to locate oil, it is probable that substantial fields of natural gas will be revealed during the search. A number of transportation systems to exploit these gas fields have been tentatively proposed including such esoteric concepts as nuclear submarine tankers shipping liquefied natural gas under the polar ice to overcome the disadvantage of the short season for conventional maritime operations along the Arctic Coast. However, the most feasible system for economic transport of the gas to industrial and domestic consumers remote from the field appears to be by pipeline South connecting with the existing Canadian gas transmission trunk lines. These comprise (a) a gas gathering system in Alberta operated by the Alberta Gas Trunk Line Company Ltd. (A.G.T.L.), and (b) a number of transmission lines transporting gas from the Alberta borders to Eastern Canada and the United States. Figure 1 shows the increase in gas transmitted by the A.G.T.L. in the 10 years following the inception of the company with a current level something in excess of 3 billion cubic feet per day (b.c.f.d.). There is no reason to suppose that the demand for gas will diminish in the foreseeable future and Figure 1 includes projection of future requirements calculated on both a pessimistic and an optimistic growth basis. Although the demand is likely to increase at a high rate, it is by no means certain that the supply from Alberta will be available to keep pace with it. This is because the Alberta Oil and Gas Conservation Board operate a licensing system that permits the export from Alberta only of that gas considered to be in excess of the reserves necessary to meet the estimated home demand for the next 30 years. The current position is that nearly all the slack in the system has been taken up so that increase in gas exports from Alberta is now largely controlled by discoveries of new reserves.

Given this situation, it is entirely possible that a steady 'base load' of 1 to 2 b.c.f.d. could be drawn from the Northern Canadian gas fields with remaining 3-4 b.c.f.d. drawn from Alberta sources in fluctuating amounts depending on the season. Figure 1 shows this load division with a 1 b.c.f.d. Arctic system operational in 1972 and supplemented by a second system operation in 1974. With continuing increase in demand and the eventual depletion of Alberta fields, it is anticipated that a progressively increasing load will be imposed on the Northern gas fields.

Northern Gas Pipeline

At the present time three different pipeline systems of 1 b.c.f.d. capacity that might be used to transport gas South from the Arctic gas fields are under study. An attempt is made here to describe in very general terms these systems but since the studies are by no means complete or comprehensive, no definitive conclusions can be drawn at this stage.

The three systems are, briefly

a) a conventional high pressure gas transmission line of welded steel pipe rated for low temperature service with natural gas fuelled turbine driven compressor stations at intervals

b) a liquefied natural gas high pressure transmission line, well insulated, and of a material suitable for cryogenic service with natural gas fuelled pumping and cooling stations at intervals

c) a non-isothermal high pressure pipeline supplied with liquefied natural gas which is progressively evaporated along the pipeline by absorption of heat from the Arctic environment to above supercritical temperatures and is delivered in the cold gaseous condition.

System (a), the conventional gas transmission line, is the standard of comparison for the other two systems and is the one system that is practicable for installation today.

System (b), the LNG line, represents an attempt to capitalise on the substantial increase in density that results from liquefaction of the gas. This could result in the use of a smaller diameter pipeline with consequent savings in cost. This advantage is offset by the need to liquefy the gas and to maintain it below the liquefaction temperature which requires the use of materials rated for cryogenic service and a well insulated pipeline. Further savings are in prospect through the use of liquid pumps rather than gas compressors but these too are diminished by the need for intermediate cooling stations.

Another important factor is that the LNG line offers a feasible alternative scheme for transport of Northern gas to Eastern Canada and U.S. markets. This involves delivery of LNG from the well areas to a Northern railhead for loading into cryogenic tank cars and rail shipment East for delivery to LNG storage reservoirs. This scheme is attractive for it relieves further load on the already strained resources of the existing gas transmission lines and increases the utilisation factor of the railway system. One further possibility receiving serious attention is that mineral concentrates in particulate form could be transported to the railhead as solids in an LNG pipeline. This could prove effective in overcoming the transportation problem which presently inhibits development of some known Northern mineral reserves.

System (c), the non-isothermal pipeline, is of interest because it removes the need to provide booster, pumping or cooling stations at intervals along the pipeline with consequent enormous saving in both capital and operational costs. This is of particular importance in such relatively remote areas as the North where the climate and topography make access very difficult. In this system all the machinery is concentrated at the wellhead or gas processing plant where a permanent base and staff would be maintained in any case. There energy is abstracted from the gas to achieve liquefaction. In subsequent passage along the pipe, heat is transferred from the environment to the liquid causing evaporation and the maintenance of pressure along the line. The heat transfer represents an enthalpy addition equivalent to an infinite number of infinitesimal pumping stations along the line. Compared with an LNG pipeline the non-isothermal system has the distinct advantages that no cooling stations are required, the insulation requirements may be considerably relaxed and that gas, not liquid, is delivered. This latter is an important feature in those situations where the gas is not required in the liquid state for storage or tank car shipment.

Comparison of Gas Transmission and LNG Pipelines

In this comparison of gas transmission and LNG pipelines a capacity of 1 b.c.f.d. was assumed. Further it was assumed the gas was 100% methane, that the maximum line pressure was 1000 lb. per sq. in. and that the pressure drop between pump stations was 300 lb. per sq. in. with the temperature of the liquid in the LNG line remaining below the saturation condition at all times. The temperature differential between cooling stations was taken as 120°F with a minimum temperature of -259°F.

Two types of insulation for the LNG line were considered, (a) evacuated perlite, (thermal conductivity k = 0.0009 BTU/hr. ft. °F pressure, 10^{-2} m.m. Hg) and (b) urethane foam (thermal conductivity k = 0.012 BTU/hr. ft. °F) in both 2 inch and 4 inch layers. A ground surface temperature of 20°F was assumed.

Both the natural gas and LNG flows were found to be in the turbulent range for pipe diameters up to 40 ins. Friction factors for the flow were calculated using the Nikuradse equation based on assumed roughness factors corresponding to (a) schedule 40 pipe for the natural gas line (b) a corrugated pipe having corrugations of 0.25 ins. depth for the LNG line. The use of corrugated pipe for the LNG line causes an increase in the friction factor of five to six times the value for smooth pipe. This is justified because of the corrugated pipe eliminates the need for special (and very expensive) contraction joints in the LNG line.

Appendix 1 contains a complete sample calculation for the distance between pumping or compressor stations for the natural gas and LNG systems. Appendix 2 contains a sample calculation for the distance between cooling stations for the LNG line.

Figure 2 is a comparison of the distance between pumping or compressor stations as a function of pipe diameter for the natural gas and LNG pipelines having a flow of 1 b.c.f.d. and a 300 lb. per sq. in. pressure drop from a maximum pressure of 1000 lb. per sq. in. Reference to Figure 2 shows that for the same pipe diameter the principal effect of the increase in density as a consequence of liquefaction is to increase the pump station spacing. For example at a pipe diameter of 40 ins. the pump station spacing increases from 150 miles for the natural gas line to 450 miles for the LNG line.

Alternatively for a given pump station spacing, say 150 miles, the required pipe diameter decreases from 40 ins. for the natural gas line to 31 ins. for the LNG line.

Figure 3 shows the distance between cooling stations for the LNG line operating with a temperature differential of 120 F between cooling stations. Curves are shown for 2 ins. and 4 ins. of urethane foam and evacuated perlite. The heat input causing the temperature increase results from two sources namely frictional degradation and thermal conduction from the environment. The two factors which principally influence the rate of heat input are thus the mass flow and the thermal conductivity of the insulation. This is evident from the equations presented in Appendix 2. It is interesting to note from Figure 3 that for the particular conditions assumed here the curves of cooling station spacing is a function of pipeline diameter attain a maximum value and then decrease. Thus for 2 ins. urethane foam the optimum cooling station spacing (about 325 miles) occurs with a pipe diameter of 27 ins. Unfortunately this pipe diameter requires a pumping station (Figure 2) about every 85 miles.

On Figure 4 the results for the pump station spacing and cooling station spacing have been plotted simultaneously. Coincidence of the spacing for the pumping and cooling stations occurs at a pipe diameter of 37 ins. and station spacing of 300 miles for the 2" Urethane foam case and for the 4" Urethane foam case at a pipe diameter of 40 ins. and a station spacing of 460 miles. Use of smaller diameter pipe would permit increased cooling station spacing with a sharply reduced pump station spacing. Very detailed economic analysis will be required to obtain the optimum system.

The results for the perlite insulated pipelines shown in Figure 3 indicate that much greater cooling station spacing can be achieved than with foam insulated lines. However the increased complexity and hence cost of evacuated lines would almost certainly make these systems uncompetitive in comparison with foam insulated lines.

Power Requirements

The power requirements for pumping are given by

$$P = \frac{\dot{m}\ \Delta P}{J\ e\ (550)}\ \text{H.P.} \qquad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \quad (1)$$

where $\dot{m}$ = mass flow rate
ΔP = pressure drop between pumping stations
ρ = mean density
e = pump efficiency

This equation applies directly for the liquid (assumed incompressible) and approximately for the gas (which is compressible). Because the density of the liquid is much greater than the density of the gas (even when compressed to 1000 lb. per sq. in. the ratio is about 7 to 1), the power requirement to pump LNG is much less than that required to pump natural gas.

However in the case of the liquid line additional power is required for the cooling engines and there is an initial power requirement to achieve the liquefaction. This latter may be minimised or even eliminated if the gas well pressure is sufficiently high to achieve liquefaction.

It is not possible to compare power requirements for the gas and liquid cases except in specific cases. It depends on the system chosen whether the power requirement for the liquid line case is greater or less than that required for the gas line.

The Evaporative Non-Isothermal Two Phase Line

This concept is intended to provide a system without pumping or cooling stations along the line so that all the machinery would be concentrated at the line terminals and on-line maintenance reduced to the absolute minimum. Thus far, a rather comprehensive theoretical analysis of the concept has been completed and the principal design parameters recognised. Currently optimum combinations of these parameters are being determined. Because of space limitations and the complexity of the analysis no attempt is made here to describe the system in detail. This will form the subject of a separate paper now in the course of preparation.

The evaporative line is seen to comprise several sections. The initial section would be a relatively small diameter line rated for cryogenic service and supplied with liquid natural gas at high pressure subcooled below the saturation temperature even to the extent of being in a 'slush' state, that is, a mixture of solid and liquid material. Slush is attractive because it increases the heat capacity by the latent heat of fusion of the solid component. It is reasonable to suppose that up to half the material could be in the solid state.

At some distance along the line the solid component has all melted and the environmental heat transfer and frictional degradation increase the temperature of the liquid. At the same time the pressure of the liquid falls. Conditions may be selected or controlled so that the critical temperature is attained while the pressure is above the critical value. In this situation the fluid changes from the liquid to the gaseous state with no appreciable change in thermodynamic properties and continues passage along the line with the gas absorbing heat and increasing in temperature while decreasing in pressure. The corresponding decrease in density and consequent increase in gas velocity may be compensated by increasing the flow section area, either by increasing the pipe diameter or installing multiple lines.

Other conditions may be chosen such that the pressure is below the critical value before the saturation temperature is attained. This situation is considerably more complex than the supercritical one described above. Further heat transfer to the saturated liquid causes progressive evaporation with the consequent substantial change in density and the establishment of two phase flow in the pipe. In process down the pipe the pressure decreases, the temperature increases. Due to the changing thermodynamic state and further environmental heat transfer and frictional degradation the vapour fraction increases until the saturated vapour state is reached. The flow now reverts to a single phase system in the gaseous state. As before, increase in the flow section area at intervals is advisable to maintain the velocity at a reasonable value.

The two phase evaporative system has been the principal subject of the theoretical analysis referred to above. Sufficient has been accomplished to demonstrate the concept is theoretically feasible. It now remains to investigate the system experimentally and determine the data somewhat arbitrarily assumed to make the theoretical model amenable to analysis.

Optimisation of the thoretical model is now in progress and with a large number of options open for study, it is not possible at the present time to finalise design. One result for an evaporative non-isothermal line is shown in Table 1. For this study a flow of 1 b.c.f.d. over a distance of 1000 miles was assumed. The line was taken to comprise four sections each 250 miles long with equal pressure drops of 100 lb. per sq. in. in each section. Saturated liquid at 600 lb. per sq. in. was admitted to the inlet of the first section and it was found that saturated vapour at 300 lb. per sq. in. occurred near the exit of the third section. Thus single phase gaseous flow occurred in the fourth section with delivery of superheated gas at 200 lb. per sq. in. For comparison the equivalent capacity line having a uniform pipe section for the whole distance as included in Table 1.

The results shown in Table 1 are perhaps not typical of prospective practice because of the very long length, 1000 miles, and the relatively low pressure regime adopted, 600-200 lb. per sq. in. However, the results do serve to demonstrate in dramatic fashion, the high price (in terms of the large pipe diameters and number of parallel pipes) that must be paid in order to dispense entirely with on line pump stations. For high pressure systems transporting over shorter distances the required pipe diameters are much more practicable. This leads to a consideration of hybrid systems with relatively widely spaced pumping/cooling stations of substantial capacity and moderate pipe diameters.

CONCLUSIONS

Substantial theoretical and experimental research on liquid natural gas pipelines is required before their economic status in comparison with natural gas transmission lines can be established.

The low temperature environment, the difficulty of access and the relatively long distances involved make the study of LNG pipelines of importance to the development of Northern gas fields.

ACKNOWLEDGEMENT

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APPENDIX 1 - Calculation of Pump and Compressor Station Spacing

The Darcy Formula was used for both the liquid and gas lines since the pressure drop (300 lbs. per sq. in.) was less than 40 per cent of the maximum pressure (1000 lb. per sq. in.). For the gas line a mean specific volume based on the inlet and outlet conditions was used.

From the momentum equation

$$\Delta P = f\rho \cdot \frac{L}{D}\,\frac{V^2}{2g_c} + \frac{\rho V \,.(V_2 - V_1)}{g_c} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \quad (1)$$

where ΔP = pressure drop

f = friction factor

L = pipe length

D = pipe diameter

ρ = mean fluid density $\left(\frac{\rho_1 + \rho_2}{2}\right)$

V = mean fluid velocity

V_2 = exit velocity

V_1 = inlet velocity

g_c = Gravitational constant

Neglecting the second term in equation (1)

$$L = \frac{\Delta P \cdot 2g_c \cdot D}{f \cdot \rho \cdot V^2}$$

$$= \frac{\pi^2 g_c \cdot \Delta P \cdot \rho \cdot D^5}{8(\dot{m})^2 f \cdot} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \quad (2)$$

The friction factor 'f' for gas lines was determined using the Moody diagram (Reference 2) showing 'f' as a function of Reynolds number Re for various pipe roughnesses. Roughness factors equivalent to schedule 40 pipe was assumed for the gas line.

Friction factors for the liquid line were calculated using the Nikuradse formula (Reference 3) where

$$1/f = [2 \log (D/K) + 1.14]^2 \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \quad (3)$$

where k = roughness height

Corrugated pipe having a mean corrugation height (equal to k) of 0.25 ins. was assumed for the liquid line.

Reynolds numbers for both the gas and liquid lines were calculated from the relation

$$Re = \frac{\rho VD}{\mu} = \frac{4\,\dot{m}}{\pi D\mu} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \quad (4)$$

where μ = fluid viscosity,

and $\dot{m}$ is the mass flow rate

The following is an example calculation for a 20 inch internal diameter line carrying 1 b.s.c.f.d. (490 lb_m/sec) of methane with a pressure drop of 300 p.s.i. from an initial pressure of 1000 p.s.i.

For a natural gas line $\mu = 1.15 \times 10^{-4}$ poise so that

$$Re = \frac{4 \times 490 \times 3600}{\mu \times 1.666 \times 115 \times 10^{-6} \times 241.91} = 4.84 \times 10^{7}$$

and then from reference (3) $f = 0.0118$

The specific volume at inlet (1000 lb per sq. in. 60°F.) = 0.31 ft^3/lb_m

at outlet (700 lb per sq. in. 60°F.) = 0.46 ft^3/lb_m

Thus, the length of pipeline between compressor stations

$$L = \frac{\pi^2 \times 32.2 \times 300 \times 144}{16 \times 490^2 \times 5280} \left(\frac{1}{0.31} + \frac{1}{0.46}\right) \frac{(1.666)^5}{0.0118} = 3.85 \text{ miles}$$

For a liquid natural gas line with f calculated from the Nikuradse equation to be 0.0408.

The length of pipeline between pumping stations

$$L = \frac{\pi^2 \times 32.2 \times 300 \times 144 \times 26.58 \times (1.666)^5}{8 \times 490^2 \times 5280 \times 0.0408} = 11.3128 \text{ miles}$$

Lengths for the natural gas line were also calculated using the General Flow Equation (Reference 4) with reasonable agreement (within 20 per cent).

From the general energy equation, neglecting kinetic and potential energy terms.

$$U_2 - U_1 = Q + p_1 \nu_1 - p_2 \nu_2 \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \text{(5)}$$

where U = internal energy
Q = heat transfer
p = pressure
ν = specific volume

For uncompressible flow, with ΔP expressed in the form of the D'Arcy equation we may write

$$\dot{m} \cdot c\ (T_2 - T_1) = \frac{\Delta T_{1m}}{\Sigma R} + \frac{\dot{m} \cdot \nu}{J}\left(f \cdot \rho \cdot \frac{L}{D} \cdot \frac{V^2}{2g_c}\right) \quad \ldots\ldots\ldots\ldots\ldots \text{(6)}$$

where $\dot{m}$ = mass flow rate
c = mean specific heat
T_2 = outlet temperature
T_1 = inlet temperature
T_o = ground surface temperature
ΔT_{1m} = log mean temperature difference of inlet, outlet and ground surface

$$= \frac{\{T_2 - T_1\} - \{T_1 - T_o\}}{\ln \left\{\dfrac{T_o - T_1}{T_o - T_2}\right\}}$$

R = resistance to heat flow
ν = specific volume
f = friction factor
ρ = density
L = length of line
D = inner diameter
V = mean velocity

Reference (5) shows that, compared with the resistance of the insulation and the soil, the resistance to heat flow of the inner pipe surface, the wall thickness and the ground surface is negligibly small.

The resistance to heat flow of the soil may be written:

$$R_{soil} = \frac{1}{2\pi \cdot k_{soil} \cdot L} \cdot \cosh^{-1} \left\{\frac{h}{R_s}\right\} = \frac{1}{2\pi \cdot k_{soil} \cdot L} \ln \left[\frac{h}{R_s} + \sqrt{\frac{h^2}{R_s{}^2} - 1} \right] \qquad \text{.............. (7)}$$

where h = centre line depth below the surface
R_s = outer radius of the insulated pipe
k_{soil} = thermal conductivity of soil

The resistance to heat flow of the insulation may be written:

$$R_{insul} = \frac{\ln. \left\{\frac{R_s}{R_o}\right\}}{2\pi.\ k_{insul} \cdot L} \qquad \text{.............................. (8)}$$

where R_s = outer radius of the insulated pipe
R_o = inner radius of the insulation
k_{insul} = thermal conductivity of the insulation

Substituting (7) and (8) in equation (6) and rearranging the cooling station spacing, L can then be expressed as:

$$L = m.c.(T_2 - T_1) \left[\frac{2\pi.\ \Delta T_{1m}}{\frac{\ln \left\{\frac{R_s}{R_o}\right\}}{k_{insul}} + \frac{1}{k_{soil}} \cdot \ln \left[\frac{h}{R_s} + \sqrt{\frac{h^2}{R_s{}^2} - 1} \right]} + \frac{\dot{m}^3\ .\ f\ .\ 16}{2g_c\ .\ J\ .\ \rho^2 .\ \pi^2 .\ D^5} \right]^{-1} \qquad \text{................. (9)}$$

Example calculation for a pipeline buried at 4 ft. from surface to centre line, 20 ins. internal diameter, insulated with 2 ins. urethane foam, having a flow capacity of 1 b.c.f.d., inlet temperature -256°F., outlet temperature -140°F.

Cooling station spacing L

$$L = \frac{1.76 \times 10^{6} \times 0.9\ (256 - 140)}{5280}\left[\frac{2\pi \times 213}{\frac{1}{0.012}\ \ln .\ \frac{12}{10} + \frac{1}{0.62}\ \ln\left[\frac{(4 \times 12)}{12} + \sqrt{\frac{(4\times12)^2}{12^2} - 1}\right]} + \frac{3600 \times 490^3 \times f \times 16 \times 12^5}{2 \times 32.2 \times 778 \times 26.5^2 \times \pi^2 \times 20^5}\right]^{-1}$$

L = 233 miles

$$\text{where } f = \left[\frac{2}{2.30}\ \ln\left\{\frac{20}{0.25}\right\} + 1.141\right]^{-2}$$

Nikuradse's equation for the friction factor with a corrugated pipe having corrugations 0.25 ins. deep.

	SECTION 1 0-250 MILES 600-500 PSI TWO-PHASE GAS-LIQUID	SECTION 2 250-500 MILES 500-400 PSI TWO-PHASE GAS-LIQUID	SECTION 3 500-750 MILES 400-300 PSI TWO-PHASE GAS-LIQUID	SECTION 4 750-1000 MILES 300-200 PSI SINGLE PHASE 1 GAS	EQUIVALENT SYSTEM OF UNIFORM PIPE DIAMETER 0-1000 MILES 600-200 PSI
NO. OF PIPES IN PARALLEL	PIPE DIA. (ins.)	PIPE DIA. (ins.)	PIPE DIA. (ins.)	PIPE DIA. (ins.)	PIPE DIA. (ins.)
1	59	66	75	97	115
2	46	52	59	75	89
3	40	45	51	65	77
4	36	40	46	58	70
5	33	37	43	53	65
6	31	35	40	51	60
7	30	33	38	48	57
8	28	32	36	46	55
9	27	30	35	43	52
10	26	29	33	42	50

TABLE .1. Evaporative Non-Isothermal Pipeline 1000 miles. Inlet condition: Saturated liquid, 600 lb. per sq. in., delivery 200 lb. per sq. in. Capacity 1 b.c.f.d.

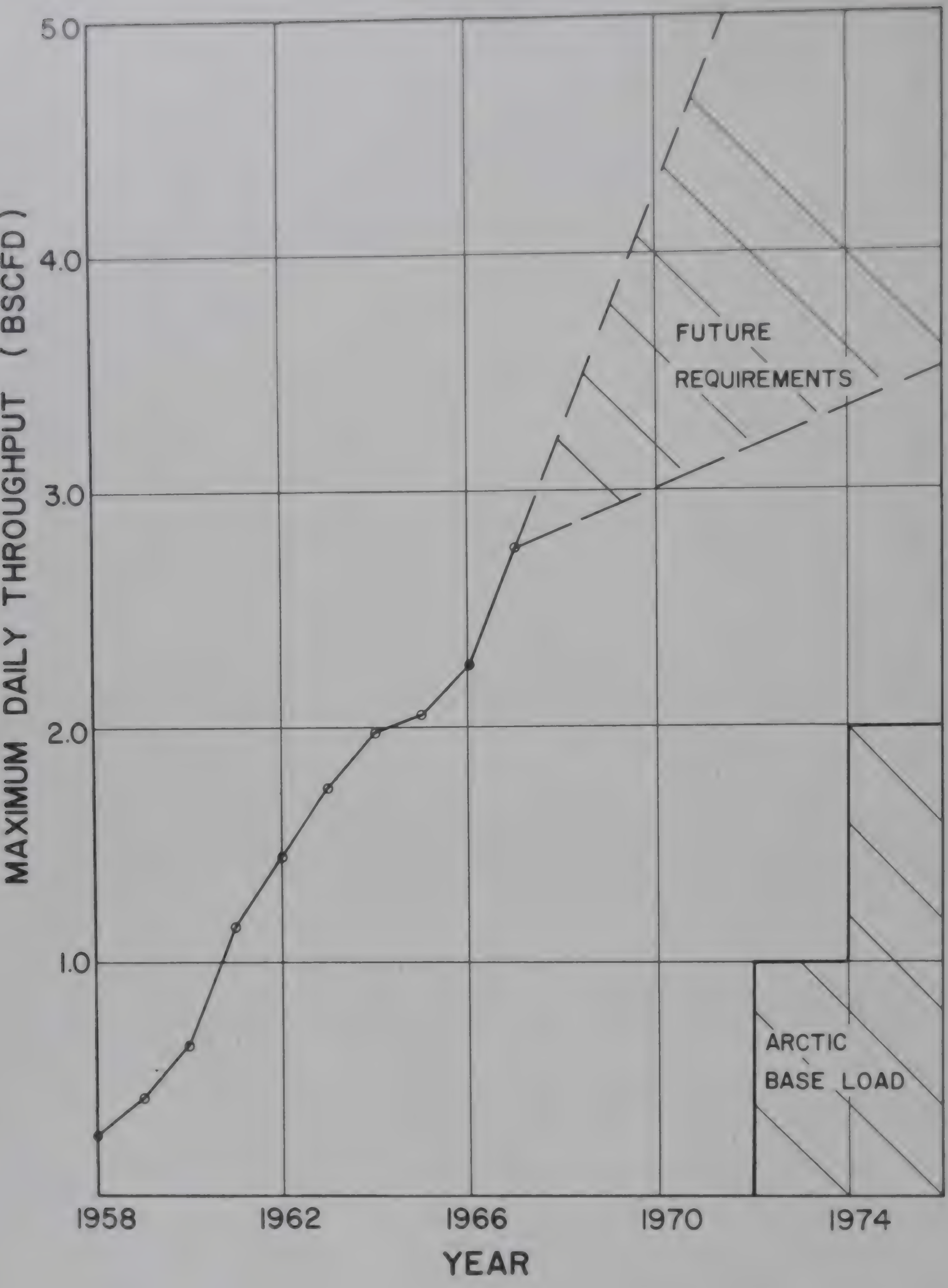

Figure 1 - Natural Gas throughput A.G.T.L. pipeline with future projections.

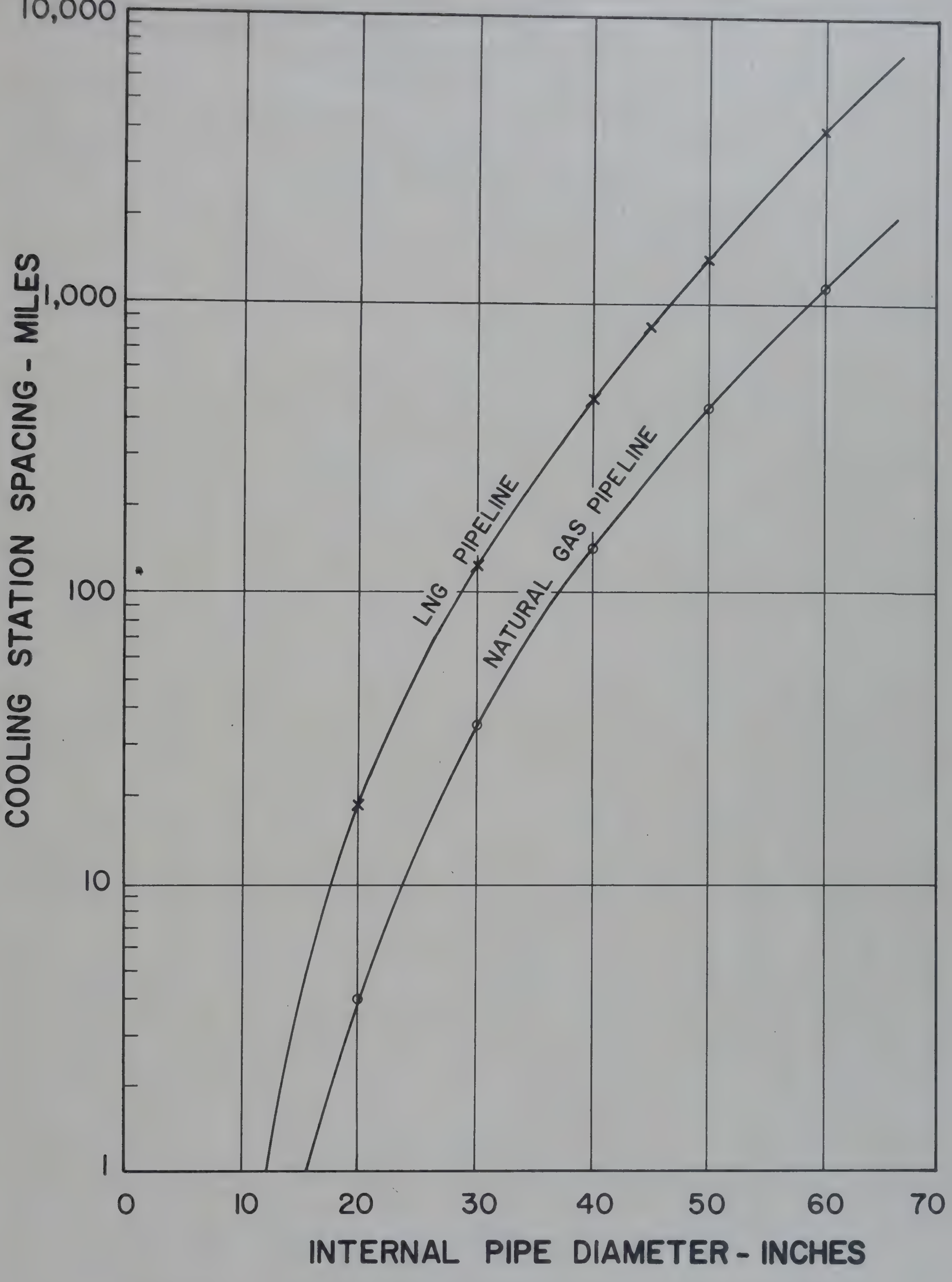

Figure 2 - Comparison of Pumping and Compressor Station Spacing for Natural Gas and LNG Pipelines.

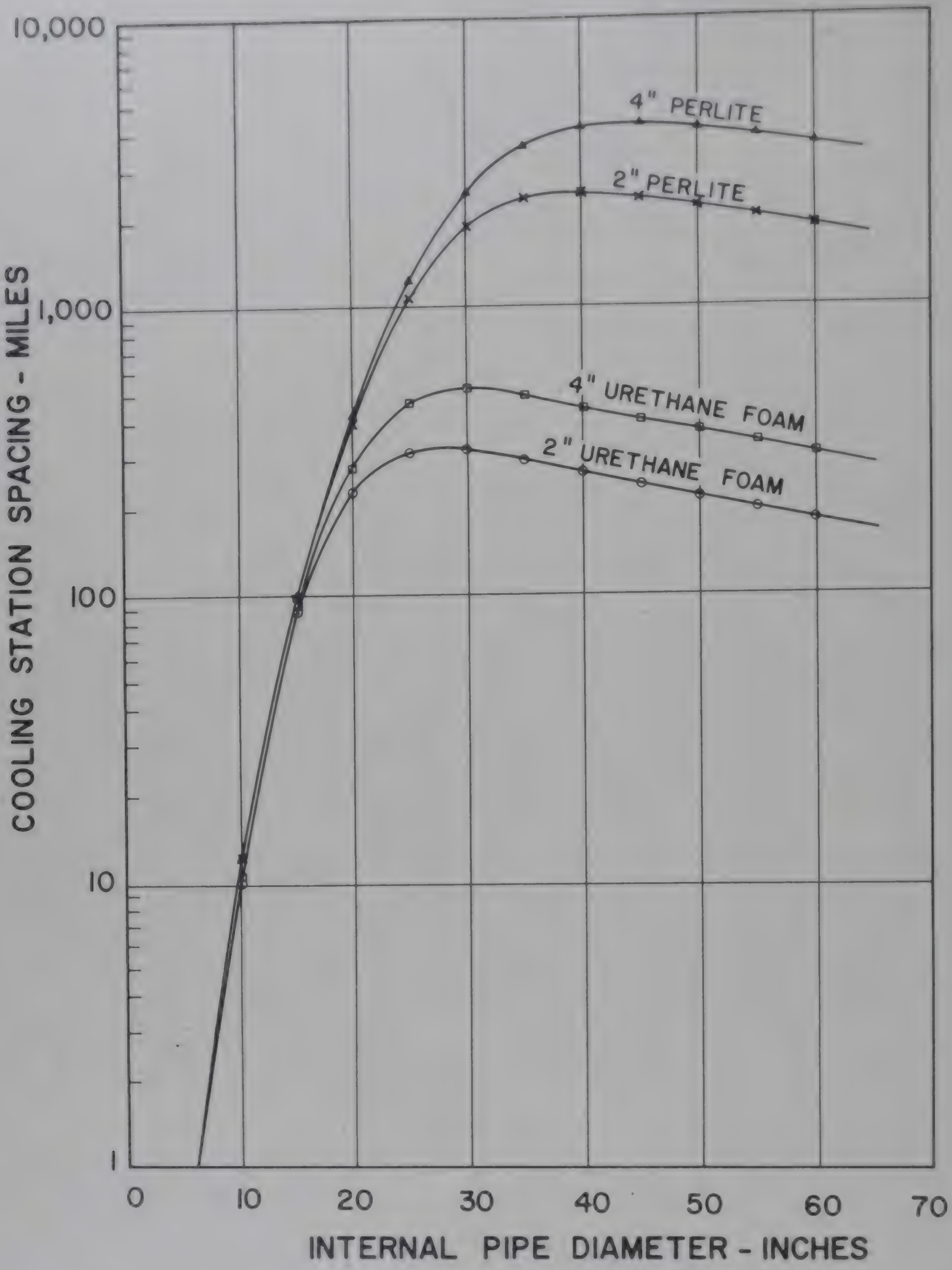

Figure 3 - Comparison of Cooling Station Spacing for LNG Pipelines.

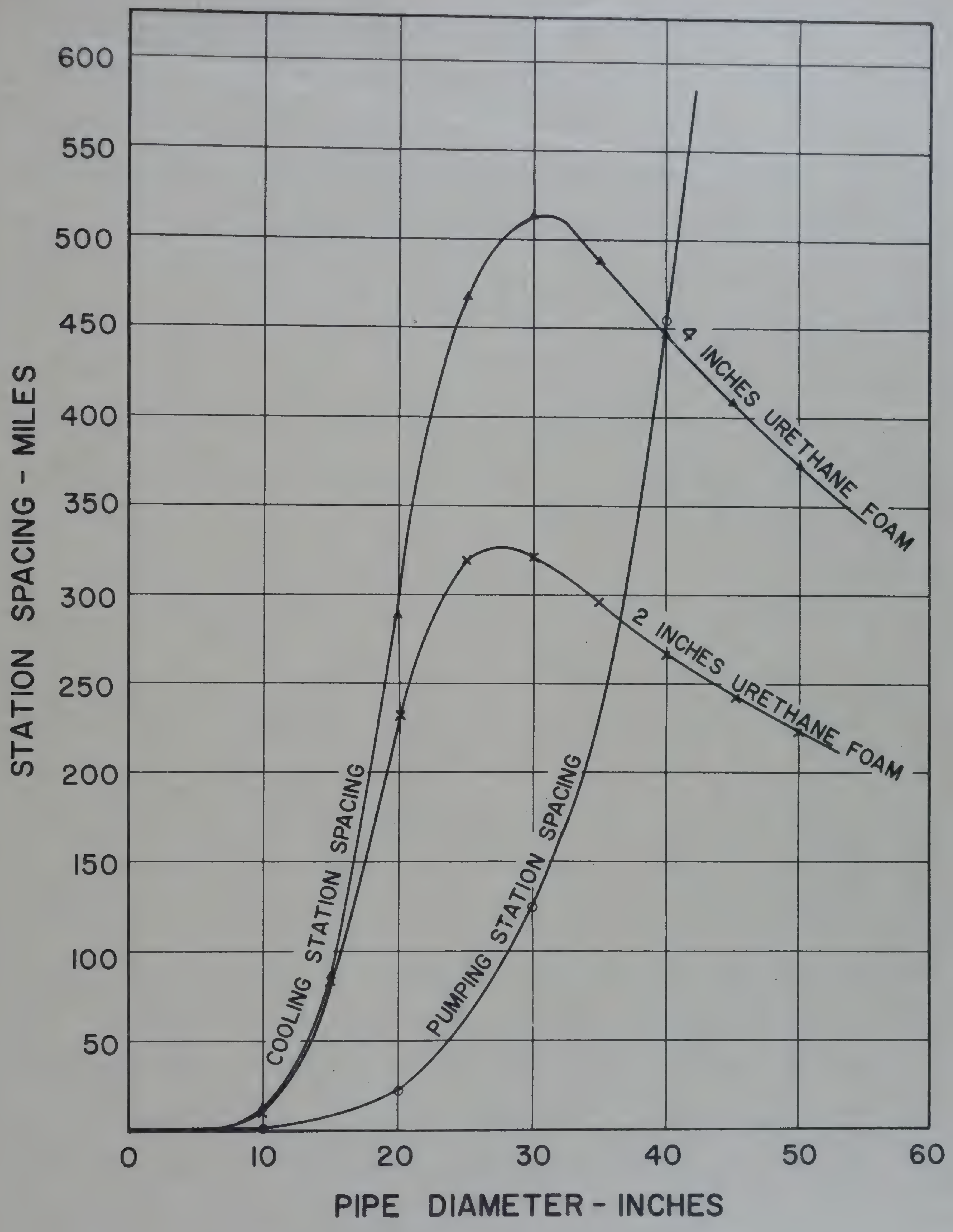

Figure 4 - Coincidence of Pump and Cooling Station Spacing for LNG Pipelines.

PROBLEMS OF LNG TRANSMISSION PIPELINES CONSTRUCTION

By O. M. IVANTSOV*

The discovery of rich gas fields in the North of West Siberia, in Central Asia and in the Yakut Autonomous Republic allows to increase highly gas production and requires other decissions in transmission of large quontities of gas over distances of 3000-6000 km.

From the Tumen fields "Urengoi" in West Siberia, the largest in the world with the reserves of 2 600 000 million m^3, from "Zapolyarnyi" field with the reserves of I 700 000 million m^3 and others it is planned in the nearest years to supply the central areas of the European part of the USSR with 100 000 million m^3 of gas annually.

The length of this pipeline will be 2500 km. The following years the amount of gas transported to this area will be doubled.

In the USSR gas pipelines of up to 1220 mm diameter are being constructed and it is planned to construct high-capacity pipelines of still larger diameters And this is the basic solution on gas transmission.

Taking into account the necessity to transport large amount of gas in one direction over vast distance it might turn out to be highly effective the natural gas transmission by mains in liquid phase, using large-capacity low-temperature (isothermal) storages, for the regulation of gas consumption.

The experience gained in the building of large-scale liquefaction plants, low-temperature (isothermal) storages and in the transportation of liquefied methane over sea, by rail tank cars and tank trucks makes it possible to change the geographic scheme of gas consumption and creats a practical base for its usage everywhere. However, there is no experience in liquefied methane transportation by pipelines over large distances. More over, to construct such mains a number of complex problems must be solved, although, one can say with certainty that LNG transportation by mains is technically feasible and represents a new logical stage of the development of large distance gas transportation.

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According to the programme and using BECM-2M computer optimal technological and constructional parameters of LNG pipeline systems (pressure, lique faction and transportation temperature, the degree of gas over-cooling, pressure drop between pumping stations, diameter, pipeline wall thickness, heat insulation thickness and etc.) were selected.

The concrete problems of LNG transportation of the volume of up to 100000 million m3 annually have been solving. The estimation showed that the optimal temperature of LNG transportation by mains is between -100°C and -120°C; the optimal pressure is about 30-40 kg/cm^2. The pipeline operation at such temperatures,the problem arises to compencate temperature shift, as well as the problem of pipeline heat insulation to prevent from loss of cold.

The maintenance of vacuum in case of vacuum insulation on large-size long distance pipelines is technically complicated and expencive, that makes practically feasible only mechanical insulation, although it is less effective than the vacuum type.

To increase effectivness and reliability of mechanical insulation and to decrease its cost, it is reasonable to use special insulating systems which provide for reduction of thermal conductiviti, i.e. the separation of plastic foarms with metallized screens, the usage of multi-ply insulation consisting of profiled plastic foam films and metallized layers,etc.

The cost of pipes is one of the main factors of the effective usage of LNG transportation by pipelines. Liquid gas pipeline capacity is approximately three to four times more than that of conventional gas pipelines,this makes it possible to reduce the diameter of the pipeline and thus to reduce the consumption of metal.

However, to construct LNG pipelines it will be necessary to use special mark of steel securing reliable operation of pipelines at low temperatures. At present, all the recomendations point to use nickel steels for low temperature pipelines, but for temperature range between -100°C and -120°C it is possible to use steels containing 3-3.5% of nickel; such steels secure the required impact value at these low temperatures.

Metallurgical research of recent years has shown that there are several promissing trends to obtain low-alloy steels, containing none or small nickel (1-1.5%). Such steels are suitable for the construction of LNG transmission pipelines.

To evaluate the possibility of using inexpensive low-alloy steels for cryogenic pipelines first of all it is necessary to define the characteristics required for the pipe material.

Besides the parameters defined by estimation on strength, resistance to brittle fracture is decisive for low temperature pipelines.

Recently a new method of parameter determination by desining was proposed in the USSR, characterising the pipeline resistance to brittle fracture. The method shows that the requirements to the appropriate properties of steel promote with the icreasement of pipeline size and stress in the walls.

According to modern theories concerning brittle fracture, brittel fracture beheviour of steel is determined by two indices - to resist a fracture initiation(characterised as the work, A_i kgm/cm^2 required for head crack initiation) and to resist a fracture propagation (characterised as the work, A_p kgm/cm^2 required for the head crack propagation).

Table 1 gives desining requirements to A_p for gas pipeline steels of various sizis. In view of responsible character of LNG pipelines it is necessary to apply checking up requirements which are accepted for sections of the 1st and 2nd categories.

Table 1

Service pressure, kg/cm^2	Diameter, mm	Wall thickness, mm	A_p kgm/cm^2 requirements of steel for 1st & 2nd section categories.
55	1220	12.5	6.6
	1020	11.0	4.9
	820	9.0	3.7
	720	8.0	3.3
	599	7.0	1.7
	529	9.0	1.1
	427	7.0	1.0
40	1220	12.5	3.5
	1020	11.0	2.6

Continued..

1	2	3	4
	820	9.0	2.0
	720	8.0	1.8
	599	7.0	0.9
	529	9.0	0.6
	427	7.0	0.5

According to the data shown in the 4th column of the table, steel for methane transmission pipe can be chosen. With internal pressure of 40 kg/cm^2, obtaining of low-alloy steel for pipe size up to 529 mm operating at even -120°C might be consider feasible. Further investigations in this field will allow to establish the possibility of using large size pipes manufactured from comparatively inexpensive steels.

For LNG pipeline A_p requirements of steel must be less in comperison to the data shown in table 1. Thus pipes of the diameter more than 529 mm may be used for LNG transmission pipelines with the level of properties required for pipes of this categopy.

It shoud be taken into account,however that even slight break is very dangerous for LNG pipelines, because a through head crack in this case can put out of action a large number of sections of the pipeline. In connection with this pipe metal for LNG transmission has to meet A_i requirements in case there are no A_p requirements. To obtain steel with high value of A_i and low A_p is more simple than to secure good resistance to fracture propagation.

In connection with the significant demand increasement for low-temperature steels, some research work are carried out in the USSR in order to obtain cheaper steels suitable for wide application.

Increasing metallurgical quolity of steel owing to the improvement of the process of deoxidation and the removal of silicon products, we managed to obtain the following results: significant temperature decreasement of the transition into brittle state of some types of steels. For example, brittle transition temperature of 35 XГ steelwas decreased up to -90 & -100°C in comparison to -60 & -70°C when conventional technology of steel production was applied.

Treatment of steel in a ladle using fluid synthetic clincer is also permited to improve greatly steel cold resistance.

For the obtaining of the low-alloy steels working at -140°C temperature other studies are being carried out concerning the development of the low-carbon non-perlit steels used after quenching and low tempering. In this case on the base of the previous studies it is possible to develop such type of steels with both the 1.5% nickei and non-nickel.

There are other methods which increase the cold resistance of the low-alloy steels. Applying the above mentioned methods to the steels alloyed with manganese, chrome, vanadium and nitrogen,it may be expected to obtain the steel plates and pipes withstanding at low temperature range from -100°C to -110°C even at the absence of the nickel and molybdenum in the steel composition. The steels withstanding low temperature range from -140°C to -150°C can be obtained by addition of the small amount of nickel /1.5%/ and possibly molybdenum /up to 0.1/.

The price of steel plates and pipes manufactured from this steel will be much less, than the price of the 3-9% nickel steel plates and pipes, and according to the approximate calculations it will be 20% more, than the cost of conventional steels, used for the structures and gas mains. It may be suggested that the low-alloy non-nickel steels with an ultimate strengh more than 50-60 kg/mm^2 can be obtained during next 3-5 years. These steels can be used for the small diameter pipes/up to 529mm/, withstanding low temperature of -120°C. The steels containing small amount of nickel /1-1.5%/ can be used for the large diameter pipes /820-1020/, working at the same conditions. In further developments it may be expected that the large diameter pipes will be manufactured with the non-nickel steel.

The base of our estimations of the strengh of the cryogenic pipelines is an ultimate state method. This method represents the receiving of the guarantee of no ultimate states of the whole structure and its separate sections will take place under full load during pipeline operation.

The tensil strength and yild strength of the steels used for the cryogenic pipelines are increasing at temperature drop.

If $R_1^H(t_{20})$ -standard tencil strength of the steel at +20°C *)

$R_2^H(t_{20})$ -standard yield strength of the steel at +20°C *)

$R_1^H(t)$ -standard tencil strength of the same steel at T°C

$R_2^H(t)$ -standard yield strength of the same steel at T°C the dependences of $\frac{R_1^H(t)}{R_1^H(t_{20})}$ and $\frac{R_2^H(t)}{R_2^H(t_{20})}$ upon ambient temperature will be as shown in Fig.1. As may be seen, with temperature drop additional reserves of bearing capacity appear together with the possibility of decreasing pipe wall thickness.

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*) +20°C is a temperature under which the mechanic properties of the steel have been determined according to the standards.

However, mechanical properties of steel /strength, yield/ even for each particular temperature are not constant. These properties are characterized by definite instability, or dispersion of strength indices. Therefore, in desining the ultimate state all possible deviations of steel strength indices from their standard values should be taken into account by introducing the uniformity coefficient and the coefficient of the material work conditions at the pipe fracture.

Thus, the design strength $R_1(t)$ and $R_2(t)$ are equal to:

$$R_1(t) = R_1^H(t).K_1.m_1;$$
$$R_2(t) = R_2^H(t).K_2 \qquad /1/$$

where K_1, K_2 - uniformity coefficients of pipe material at the steel fracture and yield strength accordingly;

m_1 - coefficient of the material work conditions at the pipe fracture.

In designing by the method of ultimate states, besides the coefficients of overloarding and of uniformity, the coefficient of work conditions of the pipeline is introduced. This coefficient allows for all the peculiarities in the work of the structure in the whole as well as of its separate elements: agressive media, stress cocentration in junctions, etc.

Thus, strength design of LNG pipelines by ultimate state method is performed according to the formula:

$$nN^{H} \leqslant \mathcal{F}\left[m; R_1(t) : R_2(t) : \ldots S\right] \quad /2/,$$

where $\mathcal{F}$ - function, corresponding to the type of load /contraction, elongation, bending, etc./,

N^{H}- stress / tencile force, bending moment/ from normal loading,

n - overload coefficient,

$R_1(t), R_2(t)$- design resistance of the pipe material,

m - pipeline work condition rate,

S - geometrical characteristics of profile.

According to this formula and considering the overload, the maximum stress in the unit must not be greater than the minimum carrying capacity of this unit. It is determined by taking into consideration the variability of material strength and pipeline work condition. Thus, designing by the ultimate state method is quite reliable.

Really, in all cases when increasing load is dangerous for pipeline strength, this method, supposing the simultaneousness of ultimate overload and minimal possible carrying capacity, may cover the most dangerous hardly probable condition.

At the section of LNG pipeline between pumping stations "n" and "n+1" /Fig.2/ Pn & Tn are product pressure and temperature at the outlet of pumping station "n", Pk & Tk are product pressure and temperature at the inlet of pumping station "n+1". As shown in Fig.2 the pressure and temperature at any section of pipeline are functions of the distance of this section from pumping station, i.e. P=P(X) and T=T(X). Then, in the designing a pipeline according to the internal pressure, expression /2/ will be:

$$\frac{nP(x)(D_н-2\delta)}{2\delta m} \leqslant R_1[t(x)]$$

or /3/

$$\frac{nP(x)(D_н-2\delta)}{2\delta m} \leqslant R_2[t(x)]$$

Equations /3/ determin the pipe wall thickness.

Not depending on the transmission regime of the product from the station "n" to the station "n+1" /Fig.2/ the pressure drop /$Pn > Pk$/ and the temperature rise /$Tn < Tk$/ of liquefied gas always take place. This can be taken into account when designing, though with the temperature rise the steel strength decreases but at the same time the pressure drops.

Assuming that the expressions $P(x), R_1[t(x)]$ and $R_2[t(x)]$ are linear, all above mentioned considerations can be represented in scheme, as shown in Fig.3. There can be two cases:

1. $R_1[t(x)] > R_2[t(x)]$ Fig.3a and
2. $R_1[t(x)] < R_2[t(x)]$ Fig.3b

If $\alpha_P < \alpha_R$ then Pk is the designed pressure, that means the inlet pressure of pumping station "n+1". When $\alpha_P > \alpha_R$ then the designed pressure is Pn - the pressure at the outlet of pumping station "n". Angles and α_P & α_R are given in Fig.3.

We can determine approximately wall thickness of the pipes for low temperature pipelines of diferent sizes at the pressure 40 kg/sm^2, if we suppose that coefficients of overload, uniformity and work conditions are the same as for conventional gas transmission lines.

The constructive design of LNG transmission pipeline is based on the need to compensate temperature displasement resulted from low operating temperature.

Table 2.

The type of transported product	Steel grade	Wall thickness,mm; size, mm			
		529	720	1020	1420
Gas P = 55 kgs/cm^2	Ct.3	7	9	13	18
	17Г C normal.	6	7	11	15
	17Г C quench.temper.	4	6	8	11
	14Г2	6	8	12	16
	10Г2 normal.	7	10	15	20
	10Г2 quench.temper.	5	6	9	12
Liquid P = 40 kgs/cm^2	06H 3	4 (3)	6(4)	8(5)	11(7)
	06H 6	4 (3)	5(3)	8(5)	10(6)
	06H 9	3(2)	5(3)	6(4)	9(6)
	X18H 9	4 (3)	6(4)	9(6)	12(9)
	Low alloy,super fine grained steel /no nickel/	3(2)	5(3)	6(4)	9(6)
	Low alloy steel of improved metallurgical quolity /no nickel/	3(2)	5(3)	6(4)	9(6)
	Low alloy, fine grained steel, improved quolity, small amount of nickel	3(2)	5(3)	6(4)	9(6)

LNG transmission pipelines may be laid either over ground onto special supports, or under ground. When laid under ground the cryogenic pipelines are thermoinsulated with hard rigid shells loosely enveloping the pipe. This ensures the possibility of longitudinal displacements of the pipe being cooled. When insulation materials are not sufficiently reliable it is necessary to use protective casing, bearing by means of compensators the weight of pipe and the weight of the ground above.

To provide for the pipeline displacements,compencating arrangements are introduced set in special cells. They are of two types: П -shaped and axial wavy. When П -shaped compensators are installed pipe consumption increases by 8-9%. Besides, П -shaped compencators creat additional pressure head losts. When using axial wavy type additional metal consumption does not exeed 5%.

Compensation of buried pipelines is more complicated than that of over land pipelines. In the Soviet Union we have the experience of gas pipeline overgrouned construction; for example, the line Taas-Tumus-Yakutsk-Pokrovsk-Bestyakh, installed on supports in the form of "zigzag" in permafrost region.

This line is succesfully operated in the temperature range of (-60°C) - (+4 0°C) and to some extent may be looked upon as a prototype of future subcooled pipelines. Fig.4b schematically illustrates temperature differences of the cryogenic and Yakutsk pipelines.

In case of LNG pipeline,its temperature difference is more and is directed to the low-temperature side., However, LNG pipeline installed at positive temperatures after being put into operation, always works at the same negative temperature,that is, it has one-sided temperature load. Yakutsk pipeline works at both negative and positive temperatures.

At lower temperatures the thermal expansion coefficient essentially decreases /Fig.5/. There fore,temperature displacements when cooling down,for instance,to -100° -160°C are about 1.5-2 times less than when heeting up to the same number of degrees. This makes it possible to lower the cost of LNG pipelines due to compencators,as compared to pipelines of Yakutsk type.

An overgrouned pipeline may be laid in plan in the form of a zigzag or in a straight line with separate sections slightly curved. The schemes of overgrouned construction are shown in Fig.6. The total length of a zigzag-shaped pipeline increases by 3-5% and of a pipeline with slightly curved sections - by 2-3%. In overland installation it is possible to use wavy compensators without extending the pipeline itself.

Spacing of the supports for the pipelines of different sizes and materials is shown in table 3.

Table 3.

	Support spacing, m	
Pipeline size, mm	Steel pipe	Aluminium alloy pipe
529	25	20
1029	32	24
1420	40	28

The specific feature of cryogenic pipelines,that is, one-sided temperature load,determines its design,which provides for prestressing the pipe by means of powerful jacks or by bending the pipe sections between dead supports in horisontal plan . /Fig.6c/. Then in cooling down these stresses would be reduced and the pipeline would take straight position and prestressing would be equal to zero. This pipe-laying method involves pipe overexertion of only 0.2-0.3%. Now this scheme seems rather hypothetic,but steel it deserves detailed study.

In overland construction method of cryogenic pipelines thermoinsulation should not be necesserily hard and of high strengh. Only at the points of support contact should the insulating layer be made of a hard material or have rigid supporting casing, as shown in Fig.6e.

The outer layer of the coating system may be wrapped by a tape or by thin aluminium.

Prelemenary analyses of technical and economical factors of under- and overgrouned laying gives the possibility to conclude that overgrouned laying of LNGtransmission pipelines is preferable, even taking into account the necessery maintenance and repair of the thermal insulation.

The gas transmission pipeline system includes storage for peak shaving. Usually storage volume amounts to 10% of annual throughput of the pipeline, i.e.the volume is very large. Unfortunately, it is not always possible for the lack of favourable geological conditions to create undergrouned storages near large gas consumption areas.

In this case the construction of large isothermal /cryogenic/ storages for LNG in the direct proximity to the consumption areas is a very favourable solution of the gas supply problem.

The problem of LNG transmission pipeline construction is technicaly feasible, but for its quick solution it is necessary to complete the research work and to construct experimental pipeline sections.

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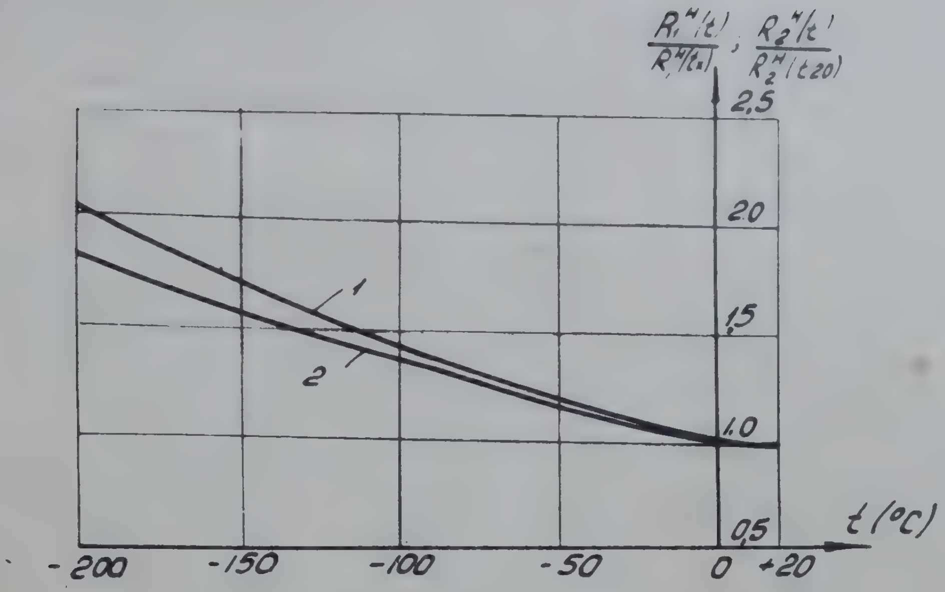

Fig.1. The dependance of steel strength characteristics upon ambient temperature.

1 - tensile strength; 2 - yield strength.

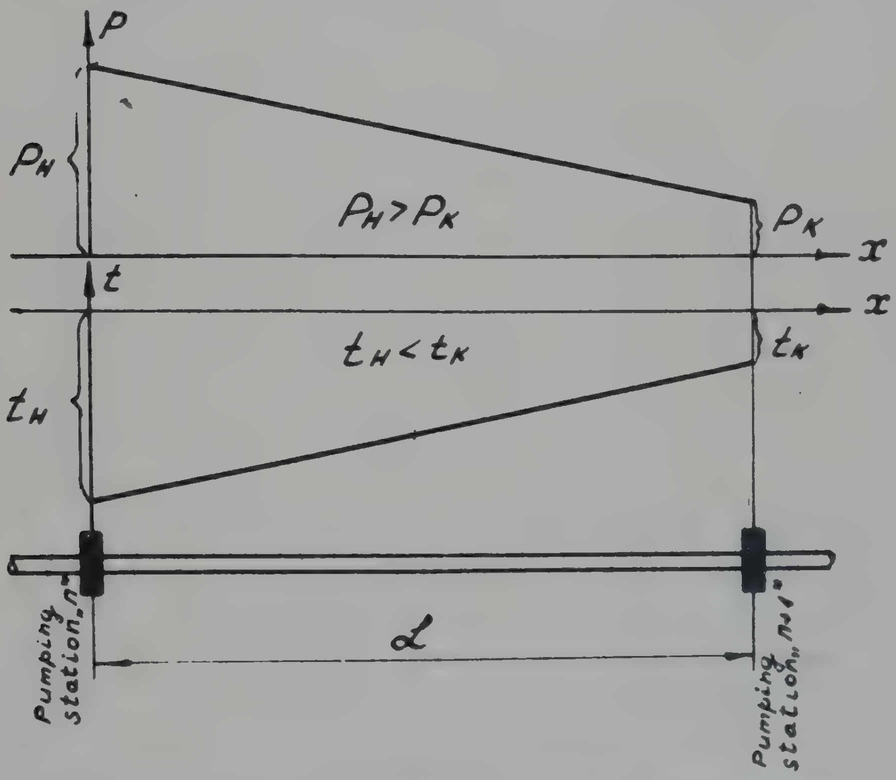
P
Pн
Pн>Pк
Pк
x
t
x
tн<tк
tк
tн
Pumping station „n"
ℒ
Pumping station „n+1"

Fig.2 The LNG pipeline section between pumping stations "n" and "n+1".

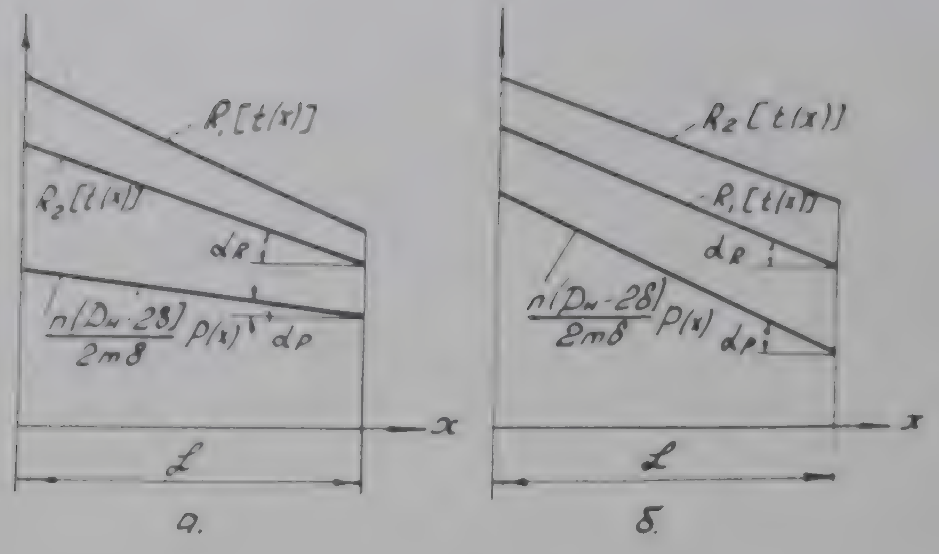
R₁[t(x)]
R₂[t(x)]
d_R
n(Dн-2δ)/2mδ P(x)
d_P
x
L
a.
R₂[t(x)]
R₁[t(x)]
d_R
n(Dн-2δ)/2mδ P(x)
d_P
x
L
б.

Fig.3 Schemes of designing strength and circular stresses variations along LNG pipeline.

a. $R_1[t(x)] > R_2[t(x)]$; b. $R_2[t(x)] < R_2[t(x)]$

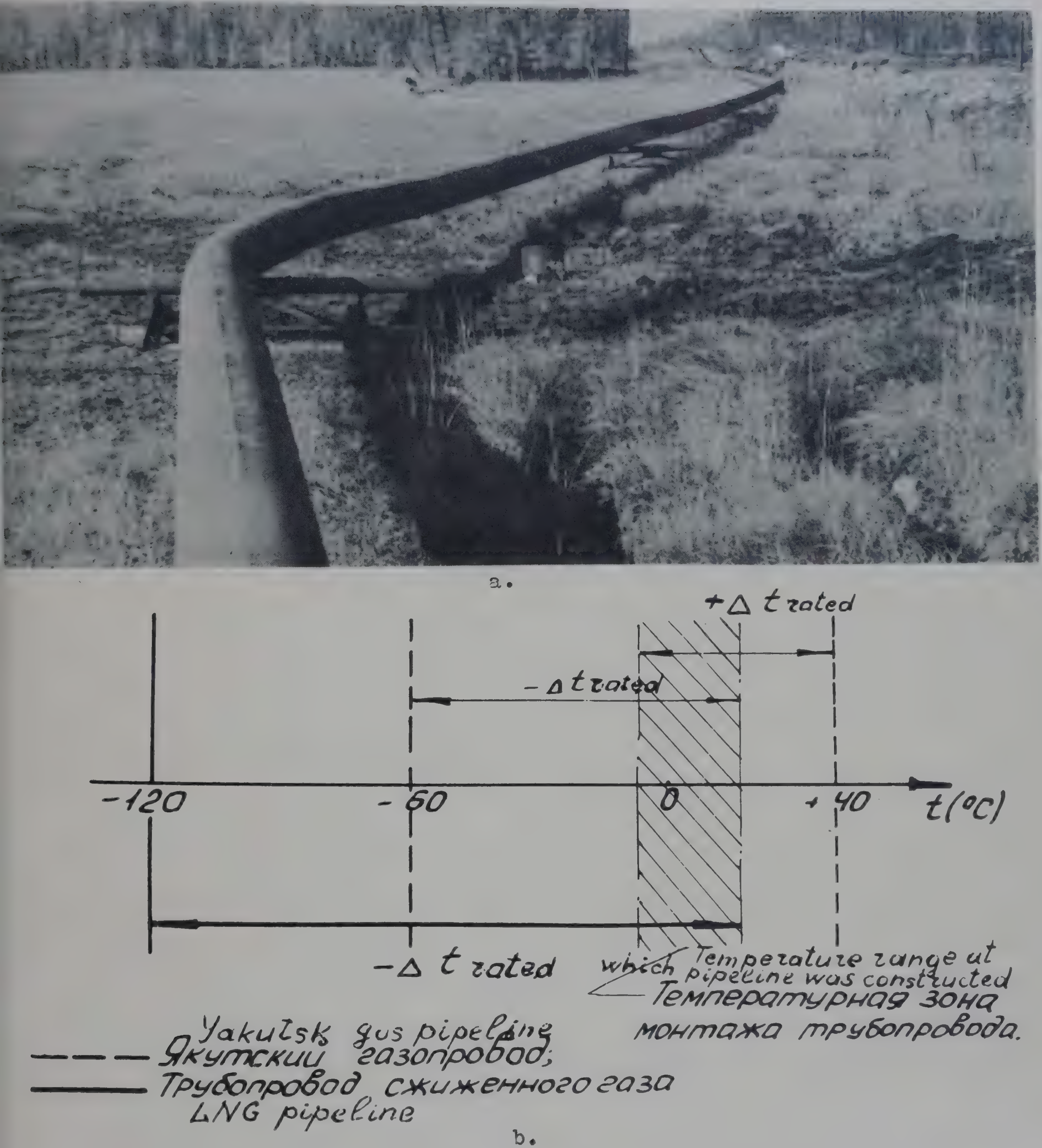
a.
+Δ t rated
- Δ t rated
-120
- 60
0
+ 40
t(°C)
-Δ t rated
which Temperature range at pipeline was constructed
Температурная зона монтажа трубопровода.
Yakutsk gas pipeline
Якутский газопровод;
Трубопровод сжиженного газа
LNG pipeline
b.

Fig.4 a. Yakutsk gas pipeline.Overgrouned laying on supports in a "zigzag" form.

b. Temperature regimes of Yakutsk gas pipeline and LNG pipeline.

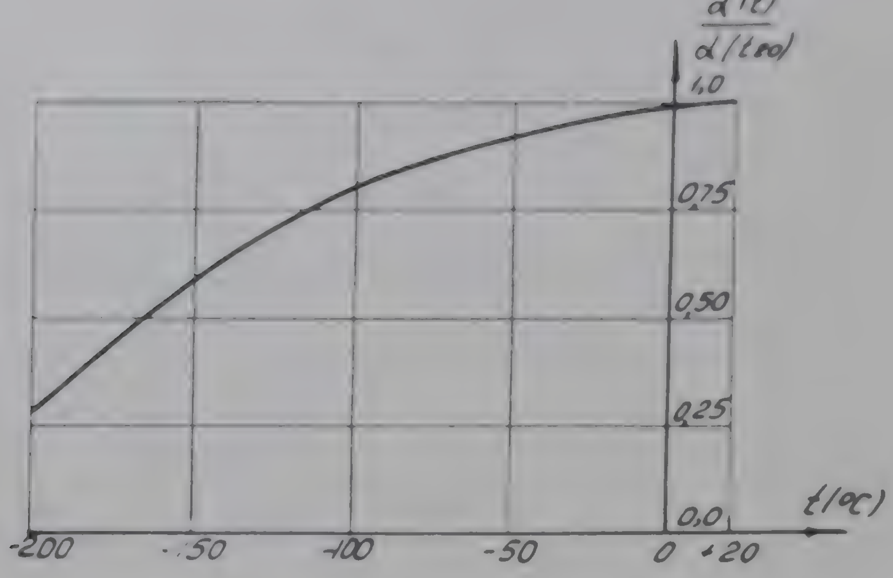

Fig. 5. The dependance of thermal expansion coefficient upon ambient temperature.

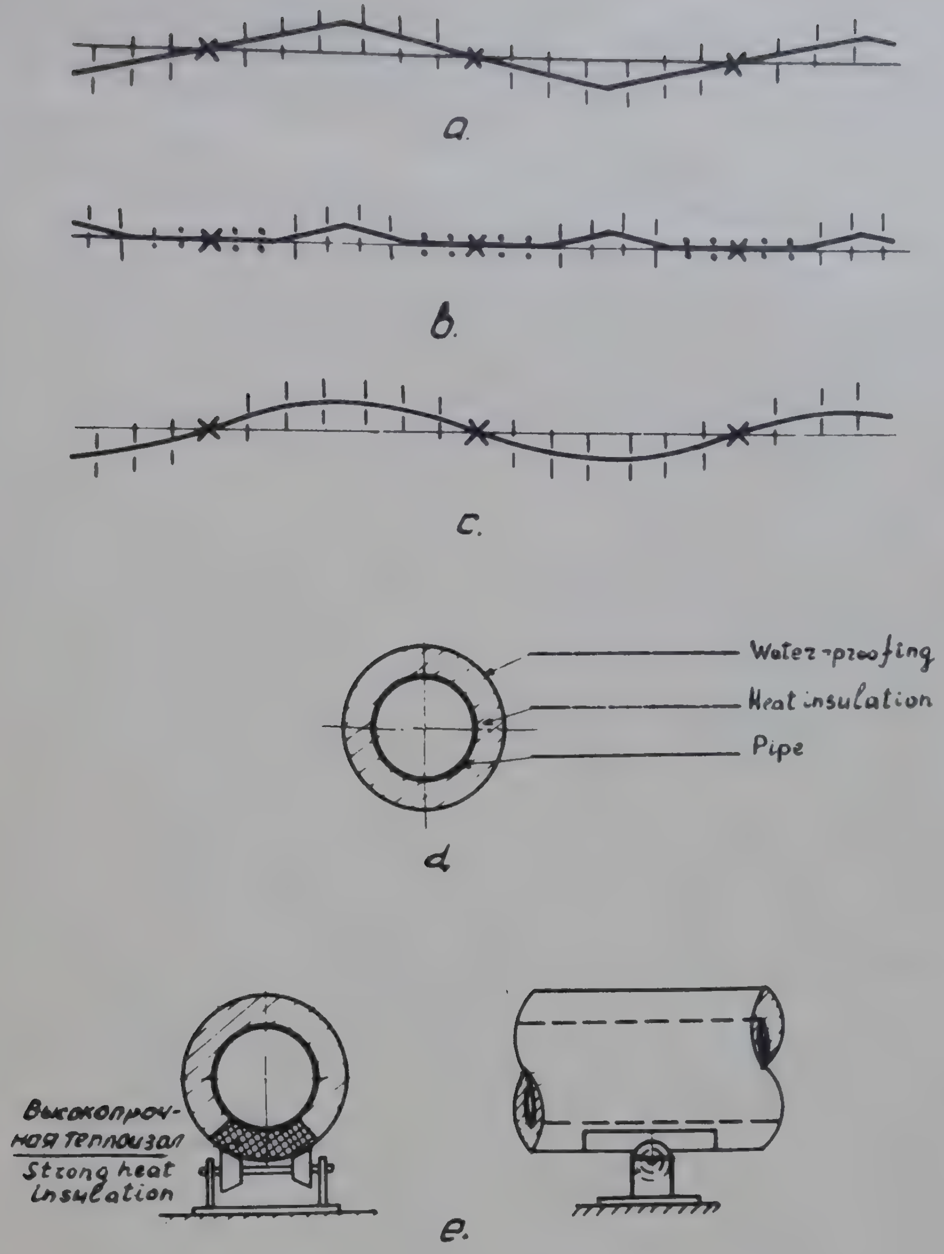
a.
b.
c.
Water-proofing
Heat insulation
Pipe
d
Высокопроч-
ная теплоизол
Strong heat
insulation
e.

Fig.6 Schemes of LNG pipeline overgroumed laying:

a) in a zigzag form;

b) with slightly curved sections;

c) with prestressing by means of bending of the pipe in horisontal plan;

d) pipeline cross section in span;

e) pipeline cross section on a support.

PIPELINE TRANSMISSION OF NATURAL GAS IN LIQUEFIED AND COOLED GASEOUS STATE

By S. F. Gudhov*

Since the need has arisen for transporting large volumes of natural gas / 100 and more thous.million cu m per annum/ over long distances, different variants of natural gas transmission by pipeline in gaseous state have been elaborated in the USSR. These are pipeline transmission by 2.0-2.5 m diameter pipes at a pressure of 55 kg/sq.cm and transmission by pipes 1.2-1.4 m in diameter at a higher pressure /100-120 kg/sq.cm/.

At the same time other practical methods of transporting large volumes of natural gas are being elaborated, namely the transportation of large volumes of natural gas in the liquefied or cooled /gaseous/ state.

1. Transmission of Liquefied Natural Gas /LNG/

The general pattern for the transmission of LNG may be presented as follows:

Natural gas from the fields is fed into the liquefaction plant and is further pumped into the pipeline.

The heat gained by LNG while transmitted by pipeline and pumpe by the pumping stations should be compensated by additional subcool ing of LNG, for which purpose subcooling units are installed at various points along the line.

At terminals evaporation installations are erected to transfor liquefied gas into gaseous state to be delivered to consumers. LNG storage facilities should be available at terminals and near big gas consumer centres.

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A characteristic feature of transmitting liquefied natural gas is a very high density of the fluid transmitted, which provides for a high capacity of the line. The density of LNG is 10 to 15 times higher than that of compressed gas in conventional conditions of natural gas transmission.

Fig.1 shows the relationship between the capacity of the line and the diameter of the pipes when LNG is transmitted. In the calculations the distance between the pumping stations was taken at 100 km, suction pressure of LNG at 30 kg/sq.cm and discharge pressure at 55 kg/sq.cm.

As seen from the diagram, a pipeline 1220 mm in diameter will carry 55 thous.million cu .m. of LNG per annum and a 1420 mm diameter pipeline will carry 80 thous. million cu .m. of gas per annum, which 3.5-3.7 times higher if compared with the conventional transmission of natural gas by pipelines of the same size with a discharge pressure of 55 kg/sq.cm.

The advantages of transporting natural gas in liquefied condition as compared to gaseous condition are as follows:

1. Overall metal investment, considering metal consumption for a liquefaction unit, subcooling units and pumping stations depending on the suction and discharge pressure values of LNG is 2 to 4 times lower than in the case of transporting natural gas in the gaseous state by a 2,500 mm diameter pipeline. With a pipeline 2.0 to 2.5 thous. km long will give a saving of 2-3 million tons of steel.

2. The transmission of natural gas in liquefied state ensures a comparatively easy solution to the problem of guaranteeing a stable gas supply and makes it possible to cope with seasonal, day and other irregularities in gas consumption by providing storage facilities for LNG in the immediate neighbourhood of large consumers.

3. In a number of cases combined extraction from natural gas of heavy hydrocarbons and helium may be carried out at the liquefaction plants.

4. A supply of gas to consumers along the pipeline route may be organized by transporting LNG in special tanks, tank-trucks or tank-cars.

5. The pipeline carrying LNG will be corrosion proof, since it will be operating at a temperature which practically excludes any development of corrosion.

The transmission of natural gas in the liquefied state should also be regarded as transmission of power contained in liquefied gas which could be utilized in consumption areas either as a refrigerant, or by converting it into other kinds of power, for instance, into the potential power of gas pressure used for a further transmission of gas, or into electric power, etc. Part of the power used for liquefying purposes could in this way be recuperated.

A practical implementation of LNG transmission will require the solution of a number of complicated problems connected, in the first place, with a considerable increase in the volume of natural gas liquefied and transmitted.

Liquefaction and subcooling plants require the use of compressor and heat-exchange equipment and are characterized by a large power load needed for liquefaction and subcooling purposes. The technological and economic indices of the units are defined in a great measure by the efficiency of the refrigeration cycle of a unit.

The minimum power consumption for refrigeration and liquefaction increases with the lowering of the ultimate cooling temperature of the fluid and decreases with the lowering of the temperature of the environment.

The actual power consumption for refrigeration and liquefaction is twice or three times higher than the minimum. The thermodynamic efficiency of the refrigeration cycles depends on the cycle chosen, but will, as a general rule, tend to fall with a lowering of the temperature. For temperatures ranging from -70°C to -150°C it will vary from 0.5 to 0.3.

The optimum transport conditions for LNG consisting mainly of methane are of the order of -120°C and -110°C.

The output of LNG by a liquefaction unit at a temperature of -115 to -125°C will be 30-35 per cent less costly than if LNG were produced at a pressure of 1 kg/sq.cm and a temperature of -161°C.

The smallest power load is achieved when cascade liquefaction cycles are used.

Different kinds of cascade cycles are used in the liquefaction plants for the output of liquefied natural gas with a pressure of 1 kg/sq.cm and a temperature of -160°C.

a) A three-cascade cycle:ammonia (or propane) - ethylene - natural gas.

b) A four-cascade cycle: propane - ethylene - methane - natural gas.

An ethylene cycle makes it possible to cool the fluid to -90°C. For further refrigeration either a methane cycle or a natural gas cycle are required.

The application of an ethane cycle (instead of an ethylene cycle) or a methane-ethane cycle will diminish the efficiency of the unit.

Refrigeration installations working on a cascade cycle require a regular supply of pure refrigerants (propane, ethylene or ethane, and methane) or else production of these refrigerants should be organized at special units within the plant.

Also applicable is a variant of the so called single-flow cascade cycle, at one time proposed by A.P.Klimenko. One refrigerant is used which consists of a mixture of hydrocarbons and nitrogen, the latter is added in case temperatures lower than -150°C are required.

The refrigerant is compressed until the pressure reaches 30-40 kg/sq.cm and is fed into the heat exchangers. The liquefied fractions are gradually separated in separators following the heat exchangers, are throttled down to a lower pressure (of about 5 kg/sq.cm) and are used as refrigerants for the cooling of both the refrigerant and the liquefied gas. Natural gas undergoing liquefaction, after having been compressed, purified of acid gases and dehydrated, is fed into the unit, cooled and liquefied in the heat exchangers by exchanging heat with the boiling liquefied fractions of the refrigerant, with a gradual lowering of the boiling temperature.

While in a cascade cycle different types of compressors are needed for the compression of each refrigerant, here only one type of compressors is needed to compress the refrigerant. The type and size of heat exchanging equipment required is reduced in number, and communications are simplified. On the other hand rectification columns have to be included in the unit to regulate the composition of the circulating refrigerant and to make up for refrigerant loss.

According to a preliminary estimate capital investment for single-flow cascade cycle units should be 5 to 10 per cent lower than for units where a cascade cycle with different refrigerants is applied. However, units in which a single-flow cascade cycle is used are characterized by a slightly increased power load as compared with units where a cascade cycle with different refrigerants is used.

Apart from the cycles discussed above, a numerous variety of gas cooling cycles may be introduced, where expansion engines are utilized and natural gas serves as working substance. These cycles are characterized by a slightly bigger power load if compared to the cascade cycles.

The application of such cycles for high-tonnage natural gas liquefaction units will require elaborating special high-efficiency expansion engines for natural gas.

Gas liquefaction cycles with expanders may be applied when pressure difference of natural gas is utilized. For instance, when gas from the main line is fed into the low pressure network for consumers located along the line, oe when it is used as fuel gas. When no such differential pressure of natural gas occurs, liquefaction units working gas cycles will be characterized by a slightly bigger power load versus units using cascade cycles.

The technological characteristics of refrigeration plants may be improved by selecting an optimum liquefaction cycle, an optimum type of compressor drive, and also by enlarging refrigeration installations and heat-exchanging and compressor equipment.

Powerful axial and centrifugal compressors should be used to compress natural gas and refrigerants. The utilization of powerful steam or gas turbines as compressor drives should be considered.

A certain effect could be produced by the utilization of condensate heat or steam heat let out by steam turbines, or else gases let out by gas turbines for the production of cold in absorption liquefaction units.

A new technological problem arises when it comes to transmitting large volumes of LNG by pipeline. The following economic aspects should be considered:

a) steel for gas mains operating at high pressure and temperature conditions of the order of $-120^{\circ}C$ should be of the cheapest possible grade;

b) heat insulation should be economically efficient;

c) the most advantageous method of laying pipelines should be chosen.

Furthermore, powerful pumps will be needed for the pumping of LNG, also appropriate fittings for the pipelines.

At present the steel used in the USSR for pipelines operating within this range of temperatures is a 3% nickel steel. The price of this steel is twice the price of manganese steel used for main line pipe production. Research is being conducted to work out cheaper and less scarce grades of steel.

The implementation of long-range transmission of LNG depends to a great extent on the way pipeline heat insulation is accomplished.

The vacuum powder insulation used nowadays is no good for gas mains of great length both for technological and economic reasons.

Calculations carried out show that when large volumes (scores of thous.million cu m per annum) are transmitted by one pipeline, it becomes practicable to use conventional kinds of insulation of the foamed polystyrene type which has a thermal conductivity coefficient, at -120°C, of about 0.025 ccal/m.h.°C.

II. Transmission of Natural Gas in Cooled State

When transmitted in cooled condition natural gas from the fields is fed into the head refrigeration installation. The transmission of cooled natural gas along the gas main is done with the help of compressors.

To make up for the heat gained from the environment, when transmitted by pipeline and compressed in compressors, it is proposed to erect special refrigeration installations for the purpose of subcooling natural gas along the route of the pipeline.

The lovering of the temperature of the gas transmitted affect the capacity of the line. The influence increasws with the rise of pressure and becomes considerable when pressure is of the order of 105-120 kg/sq.cm. Thus with an average temperature of -70°C on the line between compressor stations and discharge pressure of 120 kg/sq.cm the capacity of the line is 3.8 to 4 times higher than when natural gas is transmitted in ordinary conditions.

Such an increase in the capacity of the line under usual temperature conditions of transmitted gas could be reached only if initial gas pressure were 200 kg/sq.cm.

Up to 100 thous. million cu m of natural gas a year can be transmitted by a pipeline 1420 mm in diameter under conditions mentioned above versus 24 thous.million cu m when natural gas is transmitted by a pipeline of the name diameter with a discharge pressure of 55 kg/sq.cm and a temperature of +20°C.

A steep rise in the capacity of the pipeline with the temperature ranging between -60°C and -70°C and discharge pressure of 80-120 kg/sq.cm is accounted for not only by low temperature and high pressure but also by a considerable /threefold/ drop in the gas compressibility factor under the above temperature and pressure conditions.

Power requirement for the compression of natural gas in the compressor and the heat released are smaller when suction temperature of gas is lowered.

When natural gas cooled to -60°C or -70°C is let out from the main line into the network part of the gas could be liquefied if pressure is lowered. Hence, a comparatively easy solution, as in the case of LNG transmission, of the problem of keeping up a stable gas supply and coping with seasonal and other irregularities in gas consumption by building storage tanks for LNG in the immediate neighbourhood of large consumers. A supply of gas to consumers along the line can be effected by organizing delivery of LNG in special tanks. Extraction of helium and heavy hydrocarbons contained in natural gas can be accomplished in the main refrigeration installations or along the pipeline route. It is also possible to recuperate the power of compressed and cooled natural gas.

The transmission of natural gas in the cooled state as well as in the liquefied state makes it possible,when the need arises, to gradually raise the capacity of the line by increasing the capacity of compressor and refrigeration stations.

III. Comparative Technological and Economic Efficiency of Natural Gas Transmission in the Usual Conditions, in the Cooled and Liquefied State

When natural gas is transmitted in the cooled or liquefied state considerable expenditures are necessitated by the construction of liquefaction and refrigeration installations. These expenditures are two times lower in the case of cooled natural gas transmission as against LNG transmission.

Specific summary expenditure for fluid transmission related to 1 million cu m per day of transmitted natural gas and 1 km length of pipeline, is of the maximum order for usual gas transmission, is considerably lower for cooled gas transmission and is the lowest for

LNG transmission. Thus the comparative technological and economic efficiency of the three kinds of natural gas transmission depends on the length of the pipeline. For short distances it is expedient to apply the usual gas transmission, for long distances - LNG transmission whereas for medium distances cooled natural gas transmission is applicable.

The comparative technological and economic efficiency of the three kinds of transmission depends on a number of variables: on the relative cost of pipeline from different metals, on heat insulation, compressor and pumping stations, liquefaction units and fluid cooling and subcooling installations along the line, on the cost of construction work, the ratio of depreciation costs, and the ratio of operating costs for separate elements of the system. For the USSR, with its vast territory, it also depends on the economic ratio for regions through which the pipeline runs. For instance, if expensive main liquefaction and cooling installations are to be located in the Northern regions for which high economic zone indices are in force, transportation of natural gas in the cooled state and, particularly, in the liquefied state is the least favourable.

In table 1 are listed the technological and economic indices calculated for one definite instance: the transmission of 100 thous. million cu m of gas per annum by a pipeline running through the Komi Autonomous Soviet Socialist Republic into the Central European part of the USSR. The line is 2,150 km long.

As shown in the table, in given conditions specific summary expenditure $S = E + 0.15\ C$, where E is the operating costs and C -

Table 1. Comparative technological and economic indices of the transmission of 100 thous.million cu m of natural gas per annum in the usual, cooled and liquegied state over a distance of L = 2,150 km

Index Nº	Indices	Transmission of natural gas in the usual conditions		Transmission of natural gas in the cooled state by pipeline 1.4 m in diameter	Transmission of natural gas in the liquefied state by pipeline 1.4 m in diameter
		Pipeline diameter 1.4 m (4 lines)	Pipeline diameter 2.5 m		
1.	Specific capital investment	1	0.88	0.88	1.15
2.	Specific operational costs	1	0.89	1.03	2
3.	Specific summary expenditure $S = E + 0.15\ C$	1	0.88	0.97	1.45
4.	Specific metal expenditure	1	0.81	0.4	0.28

capital investment, turned out to be lower for transmission of natural gas in the cooled state versus traditional gas transmission by pipelines 1420 mm in diameter and lower than the corresponding expenditure for LNG transmission, though slightly higher than that for natural gas transmission by pipelines 2,500 mm in diameter. In the case of a different price index or a different pipeline route the comparative technological and economic indices of the three kinds of natural gas transmission will be different.

The technological and economic indices of the variants of natural gas transmission in liquefied and cooled state can be considerably improved at the expense of a partial recuperation of power required in main installations for the initial liquefaction or refrigeration of natural gas.

A proper utilization of power will raise substantially the economy of the whole system. The utilization of power may be effected either directly by using the cold contained in liquefied or cooled natural gas or else by converting it into mechanical or electric power. In a number of cases it will prove expedient to apply a combined method of utilizing the power of liquefied or cooled natural gas, using it partially as cold and partially by converting it into other kinds of power.

The simplest scheme for a partial utilization of the power of LNG consists in evaporating liquefied natural gas at a pressure higher than the pressure of the system into which natural gas is to be fed folloing evaporation. The necessary rise in the pressure of LNG is achieved by means of pumps. The scheme may be applied when

LNG is fed into the gas-main which will carry it to the consumer.

Thus a substantial economy can be gained during construction and operation of network pipelines and compressor stations. A considerable saving of power required for natural gas transmission is effected when the power of cooled natural gas is utilized for the purpose of transmitting gas along distribution pipelines as compared to traditional natural gas transmission. Power saving in this case may be estimated at 70 kwh per 1,000 cu m of gas or 60 per cent of the amount of power used for the cooling of natural gas in the main installations.

A certain economy can be gained when differential pressure of natural gas in the installation is used for cold production. Natural gas is then used as fuel gas for the installation or else delivered to other consumers.

Transmission of cooled natural gas solves the problem of storage og natural gas. A considerable volume, about 50 per cent, of natural gas cooled to -60°C to -75°C can be easily liquefied without additional power requirement and fed into the storage.

When LNG is pumped from the storage located at pipeline terminals or along the line and evaporized, recuperation of LNG power should be accomplished. About 20 per cent of the total amount of natural gas transmitted is intended for the generation of electric power. In future one of the methods of electric power generation will be the use of MHD generators with a cycle based on burning natural gas with oxygenated air.

The utilization of LNG cold will entail a rise in efficiency of the cycle.

The technological and economic estimates discussed above of the variants of natural gas transmission, liquefied or cooled, do not take into account the technological and economic efficiency of setting up LNG storage in the immediate neighbourhood of large natural gas consumer centres, the extraction of a helium concentrate from natural gas, the extraction of nitrogen and heavy hydrocarbons, natural gas supply to consumers through transportation of LNG in special tanks, utilization of LNG as engine fuel and for other needs, utilization of the power of cooled and liquefied natural gas. Therefore they do not give a true reflection of the relative efficiency of each variant of natural gas transmission.

An estimation of the problem as a whole will make it possible to chose for each specific case the optimum variant and optimum parameters of natural gas transmission.

On the grounds of the above technological and economic analysis it is only too obvious that pipeline transmission of natural gas in the cooled and liquefied state deserves attention on the part of engineers and research workers.

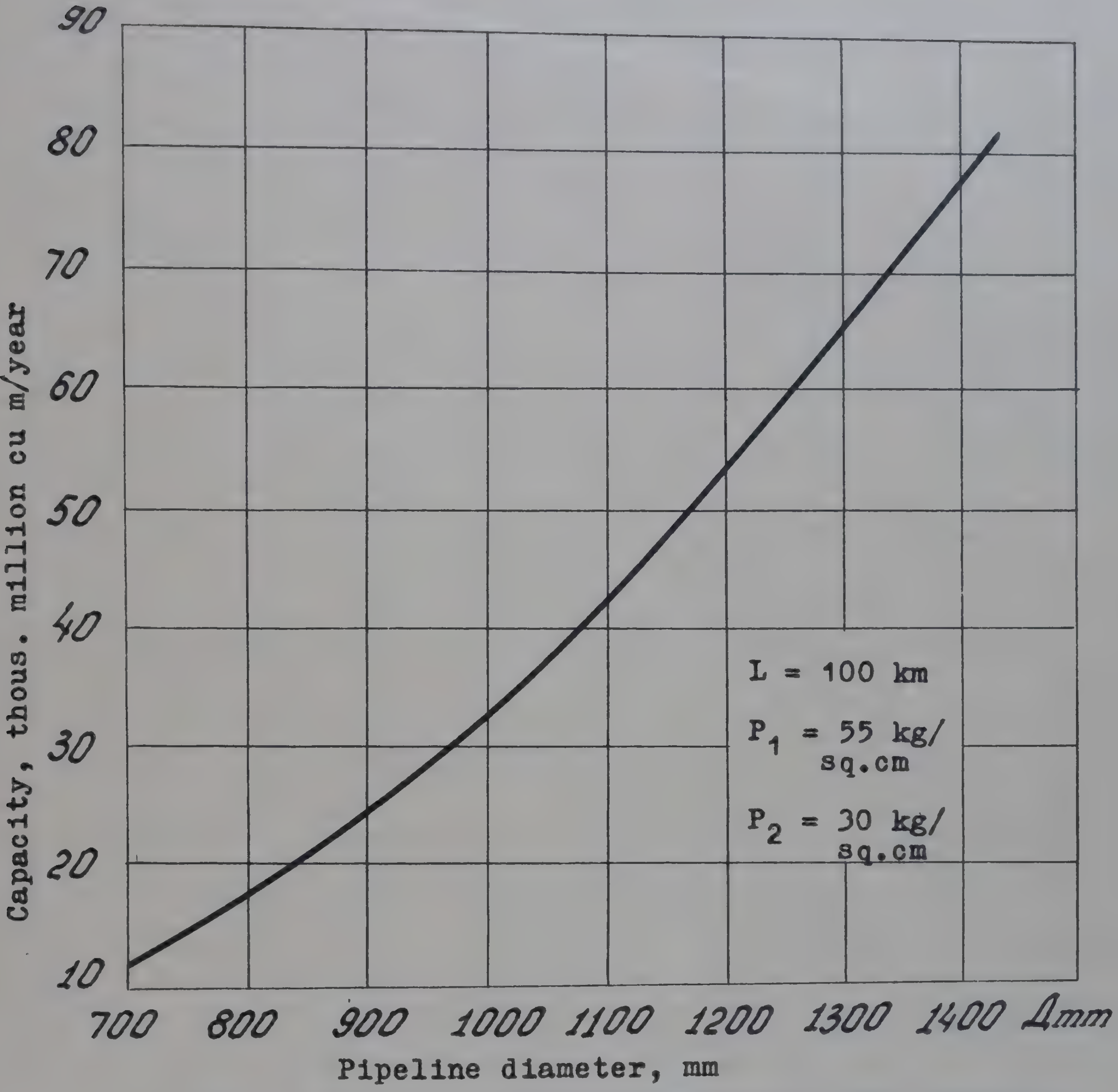

Fig.1. Relation between capacity of LNG pipeline and diameter of the pipe

TRANSPORT, TEMPORARY STORAGE AND VAPORISATION OF LIQUID NATURAL GAS

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INTRODUCTION

This paper deals with the experience obtained during the past two years in transporting Algerian Natural Gas by road tanker and vaporising into natural gas mains for the Gas Boards during their conversion programme from towns gas to natural gas.

Although possessing its own unusual characteristics, fundamentally it conforms to Cryogenic liquid handling practise, in which sphere we have considerable experience. The subject is dealt with in two sections:-

1. Road tankers and transport of LNG by road
2. Site operations

SECTION 1 : ROAD TANKERS

1. CONVERTED 3,000 GAL. VACUUM-INSULATED, SEMI-TRAILER, LIQUID OXYGEN TANKERS

The first operation with road tankers was arranged at short notice at a Gas Board's request in June 1967. For this and subsequent contracts until March 1968 3,000 gal. vacuum-insulated, semi-trailers were employed.

Before converting these tankers for LNG use the safety aspect was considered and it was decided that the bursting disc was not a suitable safety device for LNG road tankers, but should be replaced by an extra safety valve.

Pipework was modified accordingly and the safety valve exhausts extended to the roof fairing of the tanker. All flanged joints were cleaned off and "Sealogen" jointing material, which is oxygen-safe, but partially soluble in liquid hydrocarbons, was replaced by "Plasticol X10". The hose couplings were changed to the standard left-hand screwed thread with PTFE sealing ring. The flanges were bonded and an earthing block and cable fitted, having a total resistance of less than 7 ohms (Ref. 1).

Due to the low density of LNG only 6-tons could be carried instead of the normal 15-ton pay load. Another disadvantage with this type of tanker was the low working pressure, that is 30 psig.

2. PURPOSE BUILT TANKER

A tanker was designed to carry 14-tons of LNG at a maximum working pressure of 100 psig. This higher pressure facility enables the tanker to carry LNG for long journeys and have sufficient pressure capacity available to transfer the liquid into VIEs.

Fig. 1 : DIAGRAM OF 14-TON LNG TANKER

The tanker has an 8,000 gal. stainless steel tank with polyurethane insulation and aluminium weather protection. Bearers are welded to the stainless steel tank and fifth wheel and running gear attached with Tufnol heat breaks. The 3" bottom fill/liquid off-take is fitted with a vapour trap and is arranged to allow some liquid to remain in the tank after emptying - this ensures that there will always be a positive pressure in the tank for the return journey. There is also a 3" top fill line. A further 1½" liquid line and 2" vapour return are fitted for use with a pressure raising vaporiser, which is not carried on the vehicle.

TANKER FILLING

LNG is being transported from the North Thames Gas Board's terminal at Canvey and before filling, the tankers have either a residual natural gas content or are purged with nitrogen and left with a slight positive pressure. The vent gas is returned to the boil-off holder and the tankers are filled to tri-cock level - this only being opened when the tankers are nearly full, as indicated by the level gauge.

3. TRANSPORT OF LNG BY ROAD

3.1 SAFETY ASPECTS

In the absence of statutory regulations for the carriage of liquid natural gas, safe practise has been derived from standing petroleum regulations, IGE Safety Recommendations (Ref. 1) and good cryogenic practise. None of these tankers carrying natural gas is fitted with a bursting disc, but rely on safety valves of suitable size.
On the 6-ton tanker safety valves will carry full flow of vent gas, even if the vacuum fails and the pressure raising coil is left open. As an added precaution the operating wheel of the pressure raising valve is removed after use. The tankers are not vented while travelling on the road, but it is possible with the 6 ton tanker, that there will be some venting whilst the vehicle is standing overnight, due to the stratification of the liquid.

The gas venting from the safety valves is at near ambient temperature and due to its low density it diffuses rapidly into the atmosphere. An analysis of the atmosphere 5' from the vent in still air conditions indicated a methane content of less than 50 ppm.

The safety valves of 14-ton tankers were designed to take not only the normal vent rate, but also the increased vent rate should 25% of the insulation be removed. No flash arrester is fitted, but this could be added should International Regulations require this to be done.

The tankers are not parked overnight on any site which is accessible to the public. They can be parked in our own works, but then only where no oxygen plant is operating.

3.2 PRESSURE RISE DURING OPERATION

The two tankers described have very different pressure rise characteristics. The 6-ton being vacuum-insulated and 30 psig working pressure with 5% ullage space, and the 14-ton being polyurethane-insulated and 100 psig working pressure with 14% ullage space. When the tankers are moving the liquid is sufficiently mixed for it all to be at an equilibrium temperature with the vapour above. When the tankers are standing, however, stratification of the liquid will increase the rate of pressure rise.

Stratification is the formation of a layer of liquid having a higher temperature, higher vapour pressure and lower density than the main body of the liquid. The depth of this layer is effected by the turbulence of the liquid and this can vary according to:-

1. The shape of the containing vessel
2. The internal pipe arrangement
3. The ullage space
4. The surface area of the liquid

Fig. 2 : 6-TON & 14-TON TANKERS' FACTORS AFFECTING STRATIFICATION

Two tankers under consideration have substantially the same shape, but the plumbing is different. The 6-ton discharge is by dip-pipe. The only bottom connection is a 3/4" pressure raising feed. The 14-ton has a 3" and a 1½" bottom outlet and internal top fill pipe through the liquid, which will cause some agitation of the liquid and increase the depth of the stratified layer. This will decrease the rate of pressure rise in these conditions. The larger surface area and greater ullage space of the 14-ton vessel will also result in a proportionately thicker stratified layer than in the 6-ton vessel, which has a smaller surface area and smaller ullage space (5%). The following figure demonstrates the effect of this difference:-

Fig. 3 : PRESSURE RISE IN 6-TON & 14-TON LNG TANKERS

In the 6-ton tanker there is a very sharp rise of pressure when standing. Safety valve pressure being reached after ten hours, but the equilibrium pressure rise is much slower than that of the 14-ton tanker. The rise to safety valve pressure of 30 psig taking over 70 hours. The 14-ton tanker, when standing with liquid in stratified condition, takes 93 hours to reach safety valve pressure and 138 hours running in equilibrium conditions.

Fig. 4 : 14-TON LNG TANKER. PRESSURE RISE DURING 5-DAY JOURNEY

Fig. 4 shows the pressure rise in a 14-ton tanker on a 5-day delivery journey.

With overnight stops a pressure of 96 lbs is reached at the end of the fifth day and during each overnight stop there will be some stratification, but an immediate drop in pressure to equilibrium conditions when the tanker is moved on the following morning.

If the tanker is kept moving for five days with driver changeover, a pressure of 75 lbs. is reached at the end of the fifth day, but more than twice the mileage will have been covered.

It is not expected that any journey in this country will extend to more than three days. There will then be sufficient pressure capacity on arrival to be able to transfer liquid to VIEs at up to 80 psig pressure.

During the return journey the small amount of liquid left in the tanker will maintain a positive pressure. The liquid remains substantially at the bottom of the tank and as this is a stainless steel vessel with low heat conductivity the top of the tank can become quite warm. For this reason the pressure rise is much less than would normally be expected and is about 24 psi in 24 hours.

3.3 LIQUID TRANSFER RATE

The rate of liquid transfer from the tanker to VIEs or stand tanks will depend largely on the interconnecting pipework and the receiving vessel pressure. Discharge rates of up to 400 gals/min can be achieved from the 14-ton tanker.

SECTION 2 : SITE OPERATIONS

Fig. 5 : TEMPORARY NATURAL GAS SUPPLY

1. CONVERSION SITES

Most of the operations have been done at Gas Board sites feeding vaporised natural gas into towns mains during the conversion from towns gas to natural gas. In this country there are a number of outlying districts surrounding provincial towns for instance where there is only one towns gas main feeder. An external source of natural gas is required so that zones can be converted one at a time and the remainder of the town still supplied with towns gas.

Operations have varied in flow rate from 500 ft. per hour during the night on a small site, up to 100,000 cu.ft. per hour during peak periods on the larger sites. On most of these sites the flow of gas is controlled entirely by the demand and it is necessary for the regulating equipment to keep the mains pressure within the limits required at all flows.

The pressures vary from district to district, the lowest being 15" w.g. and the highest 65 psig. Control gear and vaporisers must be capable of at least 20 to 1 turn down in flow rate.

A brief description of some of the sites follows:-

2. SMALL SITES

Fig. 6 TANKER CHIMNEY VAPORISERS AND CONTROL GEAR

Fig. 6 shows a small site which operated for three weeks using 6-ton tanker and vaporising direct through air-heated vaporisers with pressure control by a conventional system of governors feeding direct into the towns mains.

The safety recommendations (Ref. 1) state that temporary storage, whether VIEs or tankers, must be surrounded by a bund wall high enough to contain all the liquid stored should there be a leak or burst vessel. On this site, however, the slope of the land and existing banks rendered this precaution unnecessary.

3. LARGE SITES

On larger sites temporary storage is in the form of vacuum-insulated evaporators using suitably-sized vessels up to a capacity of just over 16-tons, that is about 3/4 MM cu.ft. of natural gas. These VIEs are controlled at about 60 psig by the use of automatic pressure raising facilities. Suitable safety valves are fitted, the outlet being shielded and vented at some suitable height, but none of these vessels in use on natural gas have bursting discs. (Ref. 1).

Fig. 7 : PLAN - VAPORISING AND DISTRIBUTION CENTRE

4.1 COMBINED VAPORISING & DISTRIBUTION SITE

This site was of special interest, as it was used as a combined vaporising and liquid distribution centre. The site was surrounded by a 6 ft. wire fence with two sets of double gates. Two VIEs each of 16-tons capacity and operating at 60 psig pressure were installed inside a bund wall. A steam-heated vaporiser with 40,000 cu.ft./hr. capacity was also fitted into the bunded area. Two steam boilers were located outside the fence at a distance of 60 ft. from the nearest gas line. Gas pressure from the vaporiser was controlled by three governors. Two in parallel were set at 16" w.g. and 18" w.g. and the third controlled by the downstream lime pressure set at 20" w.g. This was a safeguard should either of the others fail.

14-ton tankers were used for trunking LNG from Canvey and could be used on site for filling the VIEs for local demand or for filling 6-ton tankers for use on three other conversion sites. While the 6-ton was being filled the flash gas was passed through a chimney-type ambient heat exchanger through a separate set of governors and thence to the main.

Vaporised Algerian LNG has a Wobbe index of 1430 and to adjust this to the North Sea Gas Wobbe index of 1360 air ballasting was used. An odorant was also added.

The tankers while on site were parked inside a bund wall, one side of which was built with sand bags to allow access.

4.2 On the largest site operated to date one 16-ton VIE and three 8-ton VIEs were used for storage and the maximum flow rate was 100,000 cu.ft./hr.

Fig. 8 : 14-TON TANKER FILLING VIEs AT FORDS

5. INDUSTRIAL AND EMERGENCY SUPPLIES

5.1 FACTORY SUPPLY

On this site at Fords, Basildon, the natural gas main was expected to be connected in November 1968. To avoid loss of production, the site was converted from the towns gas to natural gas by our associate Company "Seaflame" during the annual vacation in August 1968. Supplies were maintained by road tanker for four months and during that time over

5.2 SUPPLY FOR GAS ENGINE PROVING TRIALS

A supply of natural gas required for gas engine proving trials. 14-ton tankers were used through ambient vaporisers to governors controlled at 65 psig.

5.3 EMERGENCY SUPPLY

A town was supplied with vaporised LNG for two weeks when a main feeder became blocked. 2 x 8 ton VIEs were used in conjunction with an indirect heated vaporiser.

6. VAPORISERS

There are many types of vaporiser, but they can be divided into three categories.

1. Ambient Vaporisers
2. Indirect Heated Vaporisers
3. Direct Heated Vaporisers

6.1 AMBIENT VAPORISERS

There are some sites where site services are not available and some where the customer prefers to operate in this way. On these sites ambient vaporisers are the only type that can be used. These have been used up to 65,000 cu.ft. per hour. At flow rates up to 30,000 cu.ft./hr. chimney or finned tube vaporisers are used.

Fig. 9 : CHIMNEY & FINNED TUBE AMBIENT VAPORISERS

With all air-heated vaporisers, clearly the weather is the major factor. The best conditions being a hot sunny day with a gentle breeze blowing, rather Californian weather than English! The worst condition is a freezing fog i.e. English winter! It is not possible to budget for either set of conditions. In still-air conditions the atmospheric vaporiser depends on the down flow of air induced by the vaporiser being cold at the bottom and warm at the top, and for the most efficient operation a bottom inlet for the liquid is necessary into each of the finned tubes or chimneys. Where this is not done the element with a down flow of liquid loses efficiency.

Fig.10 : SPIRAL TUBE MODULES (TONNAGE OXYGEN)

There are in use in oxygen service some larger atmospheric vaporisers consisting of flat spools mounted into modules and the unit shown will vaporise a 100,000 cu.ft. of oxygen with a trimming heater, but as it is operated in three separate layers it does require some built-in pressure drop for the two lower levels to allow even distribution of the liquid.

Some of these have been modified for use with LNG, but are operated on one level to avoid the use of extra pressure drop. On LNG service, 12 modules will vaporise 65,000 cu.ft. per hour with a 10°C approach temperature.

The liquid is fed into the bottom spool of each module and is all vaporised by the end of the second spool. The vaporisers are thus self-compensating and there is no likelihood of any module flooding. As most operations entail a very high turndown, the modules are arranged in banks of four to allow sections to be isolated during slack periods. By this means it is possible to thaw out sections and have all the vaporiser ice-free for the peak periods.

FAN ASSISTED AIR-HEATED VAPORISERS

The use of fans to assist the flow of air across ambient vaporisers can increase their capacity or decrease the approach temperature (the difference between ambient temperature and the vaporised gas exit temperature). The advantage of air-heated vaporisers is their simplicity, and fan assistance adds complication in the way of flame-proof motors, switch gear and the installation of site services. The use of a fan will depend entirely on the site concerned and will not be an advantage in all cases.

6.2 INDIRECT HEATED VAPORISERS

a) STEAM HEATED WATER JACKETED VAPORISER

The steam-heated water-jacketed vaporisers used in the above operations have a four-start vaporising coil, the pressure drop with LNG being 10 psi at 30,000 cu.ft./hr.

b) EXHAUST HEAT VAPORISERS

Another version of the indirect-heated vaporiser is shown in Fig. 11. This consists of a water bath, which is heated by the exhaust gases from a natural gas burner or burners. The water is recycled over the heat exchanger coil and over a weir. Turn down could be a problem, but excess air and/or multiple burners with automatic temperature control should solve this.

6.3 DIRECT HEAT VAPORISERS

These are not recommended (Ref. 1).

7. SOME PROBLEMS ARISING FROM MIXED LIQUIDS

Fig. 12 : COMPOSITION OF ALGERIAN NATURAL GAS

7.1 Algerian natural gas has a composition shown in Fig. 12. The difference in boiling points of the constituent gases can give rise to a number of problems:-

7.2 PREFERENTIAL VAPORISATION

When a vaporiser is operating at say 30,000 cu.ft./hr. there will be a certain volume of liquid in the vaporiser. If the flow rate is reduced rapidly to say 1,500 cu.ft./hr. there will probably be sufficient liquid in the vaporiser to supply this gas flow and thus reduce the input of liquid to zero. In these circumstances methane will vaporise preferentially from the residual liquid and it was thought that the variation of the Wobbe index would be excessive. Accordingly, a test was carried out with a spiral tube vaporiser module and the results are shown in Fig. 13 and indicate that the Wobbe index remains within 2% of its original fig

7.3 RECONDENSATION OF PROPANE AND BUTANE

Some recondensation of the Propane/Butane fraction occurred in the vent line of the 6-ton tanker. This caused some weep past the safety valves, which were then repositioned to avoid any possibility of liquid remaining on the seats of the safety valves.

In view of the experience with the 6-ton tanker the vent vapour return line and safety valves on the 14-ton tanker were carefully sited to avoid recondensation.

CONCLUSIONS

Operations with LNG tankers during the past two years have proved that LNG transport by road and vaporisation on temporary or semi-permanent sites is a satisfactory method of supplying natural gas to small towns, factories etc., as either a planned or emergency operation.

REFERENCE

1. Safety recommendations
for LNG

IGE/SR/11 Parts 1 & 2

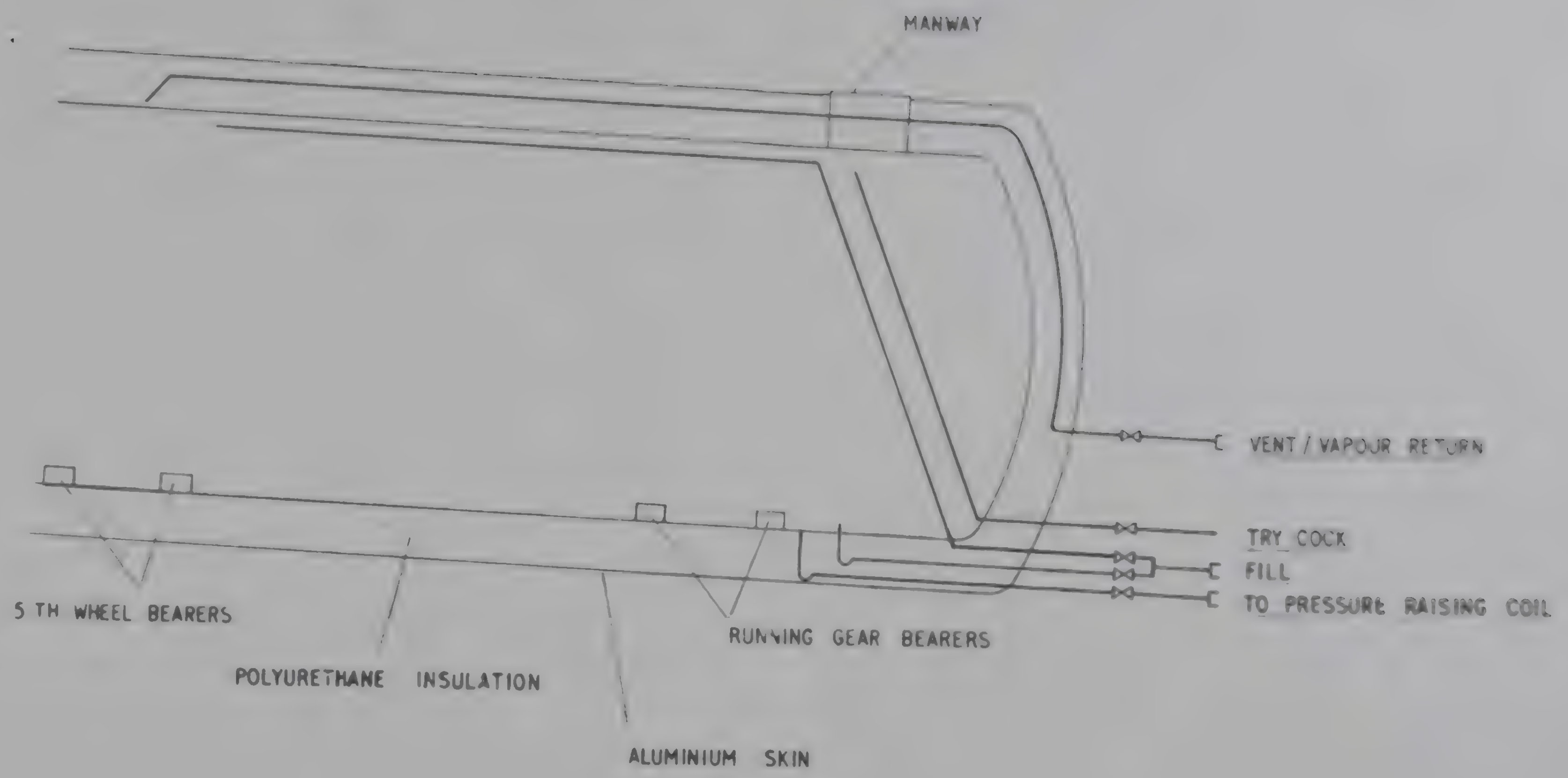
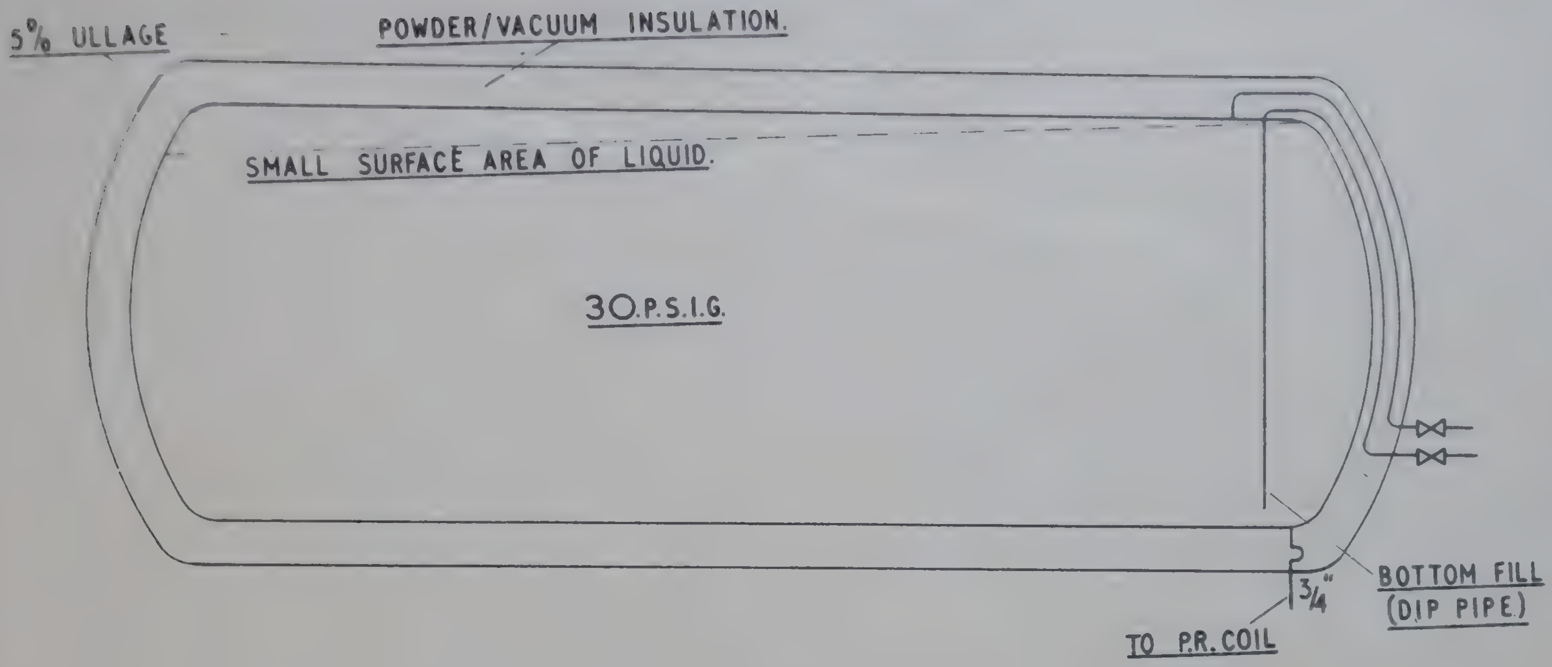
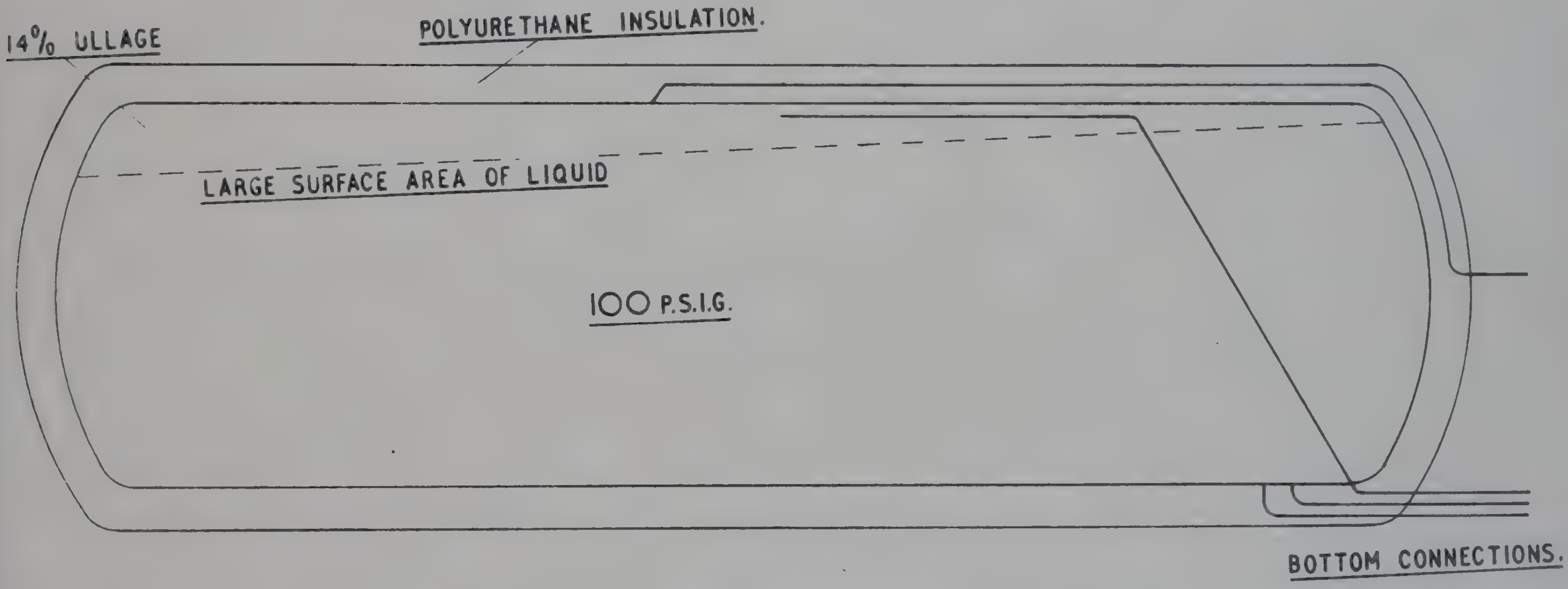

MANWAY
VENT / VAPOUR RETURN
TRY COCK
FILL
TO PRESSURE RAISING COIL
5 TH WHEEL BEARERS
RUNNING GEAR BEARERS
POLYURETHANE INSULATION
ALUMINIUM SKIN

FIG. 1. 14 TON. L.N.G. TANKER.


5% ULLAGE
POWDER/VACUUM INSULATION.
SMALL SURFACE AREA OF LIQUID.
30 P.S.I.G.
3/4"
BOTTOM FILL
(DIP PIPE)
TO P.R. COIL

6 TON (3000 GAL.) L.N.G. TANKER.


14% ULLAGE
POLYURETHANE INSULATION.
LARGE SURFACE AREA OF LIQUID
100 P.S.I.G.
BOTTOM CONNECTIONS.

14 TON (8000 GAL.) L.N.G. TANKER.

FIG. 2. 6 TON AND 14 TON L.N.G. TANKERS.
FACTORS AFFECTING STRATIFICATION.

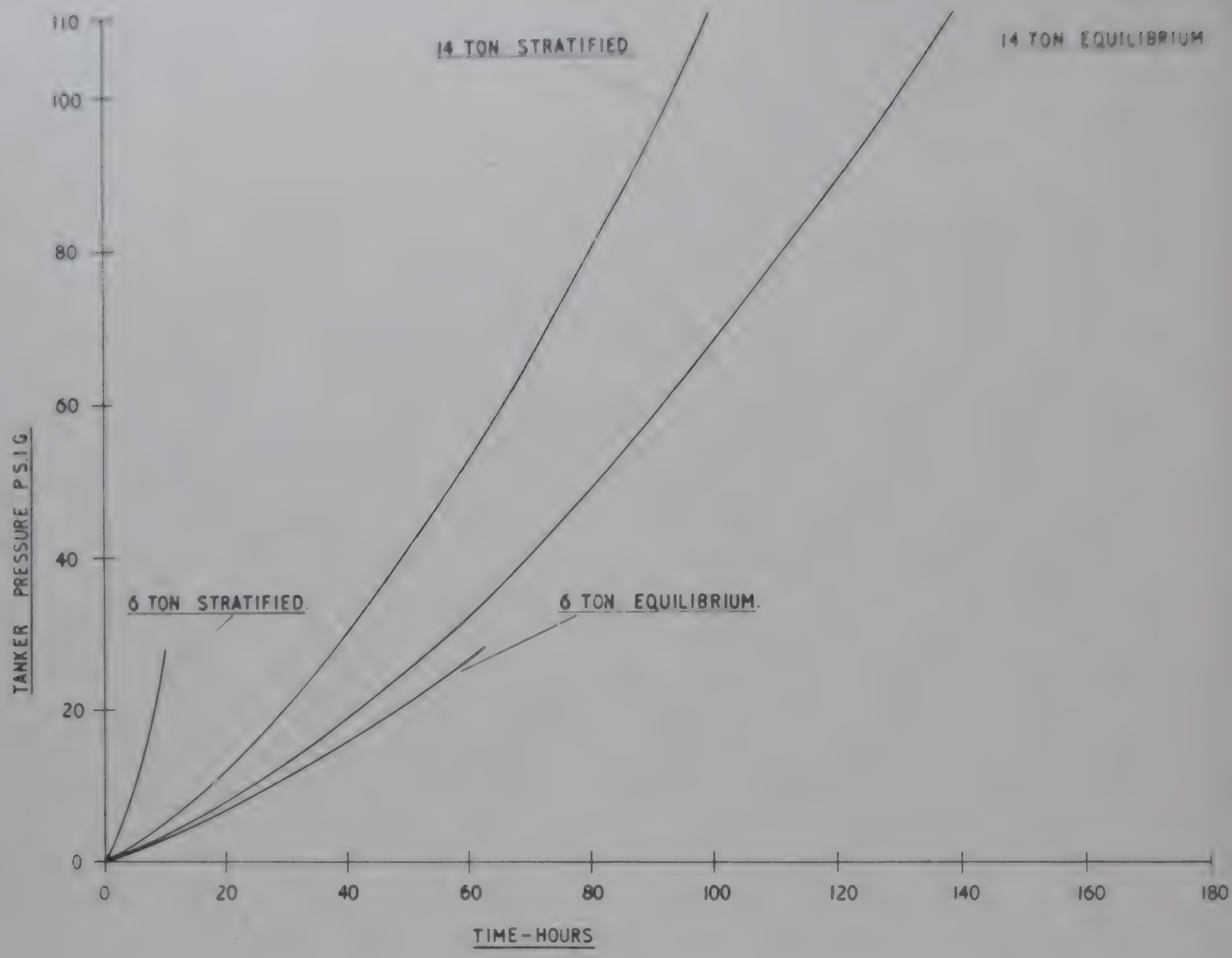


FIG. 3. 14 TON & 6 TON L.N.G TANKER PRESSURE RISE.

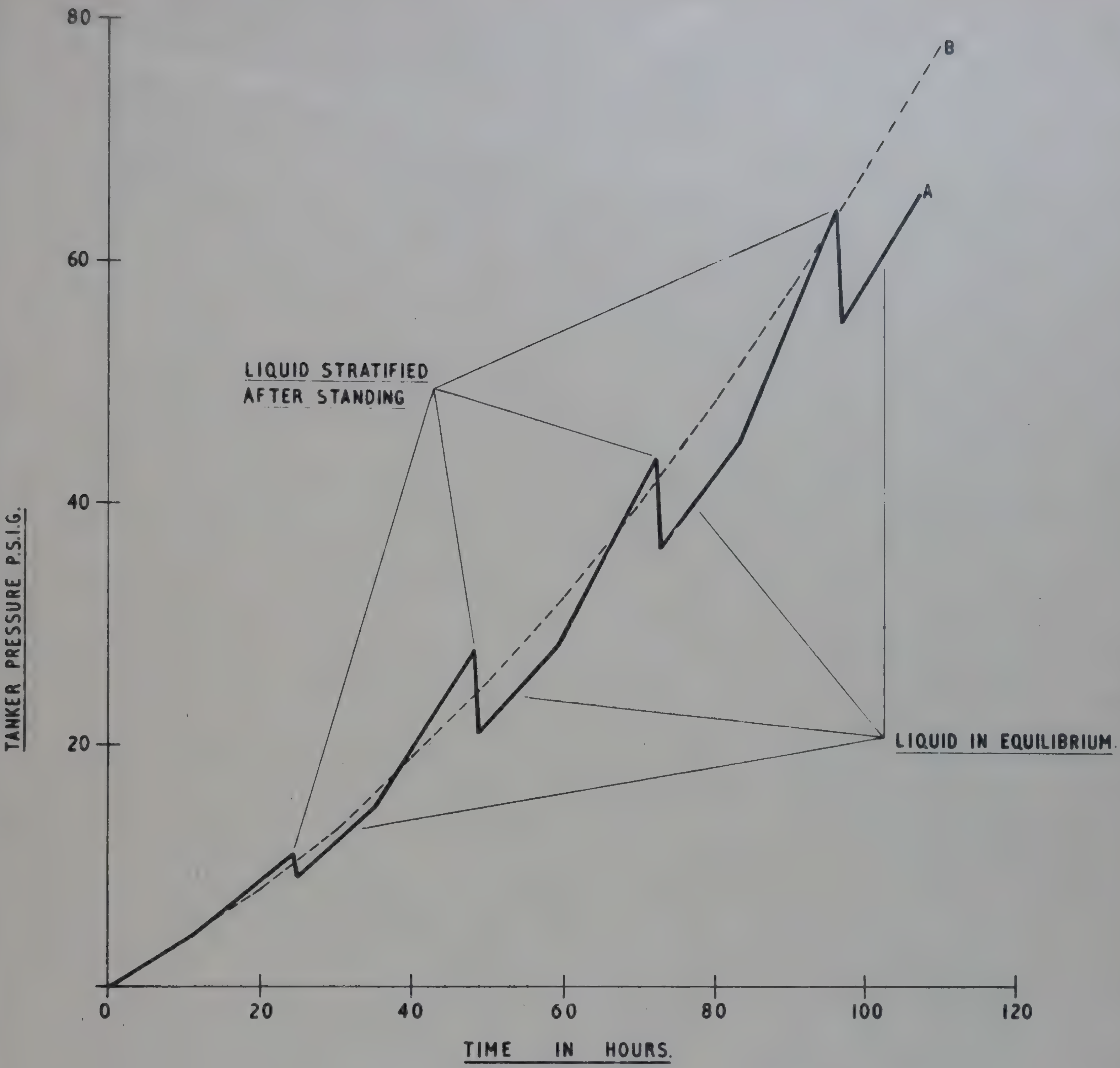


FIG.4. PRESSURE RISE IN 14TON L.N.G. TANKER DURING 5 DAY RUN.

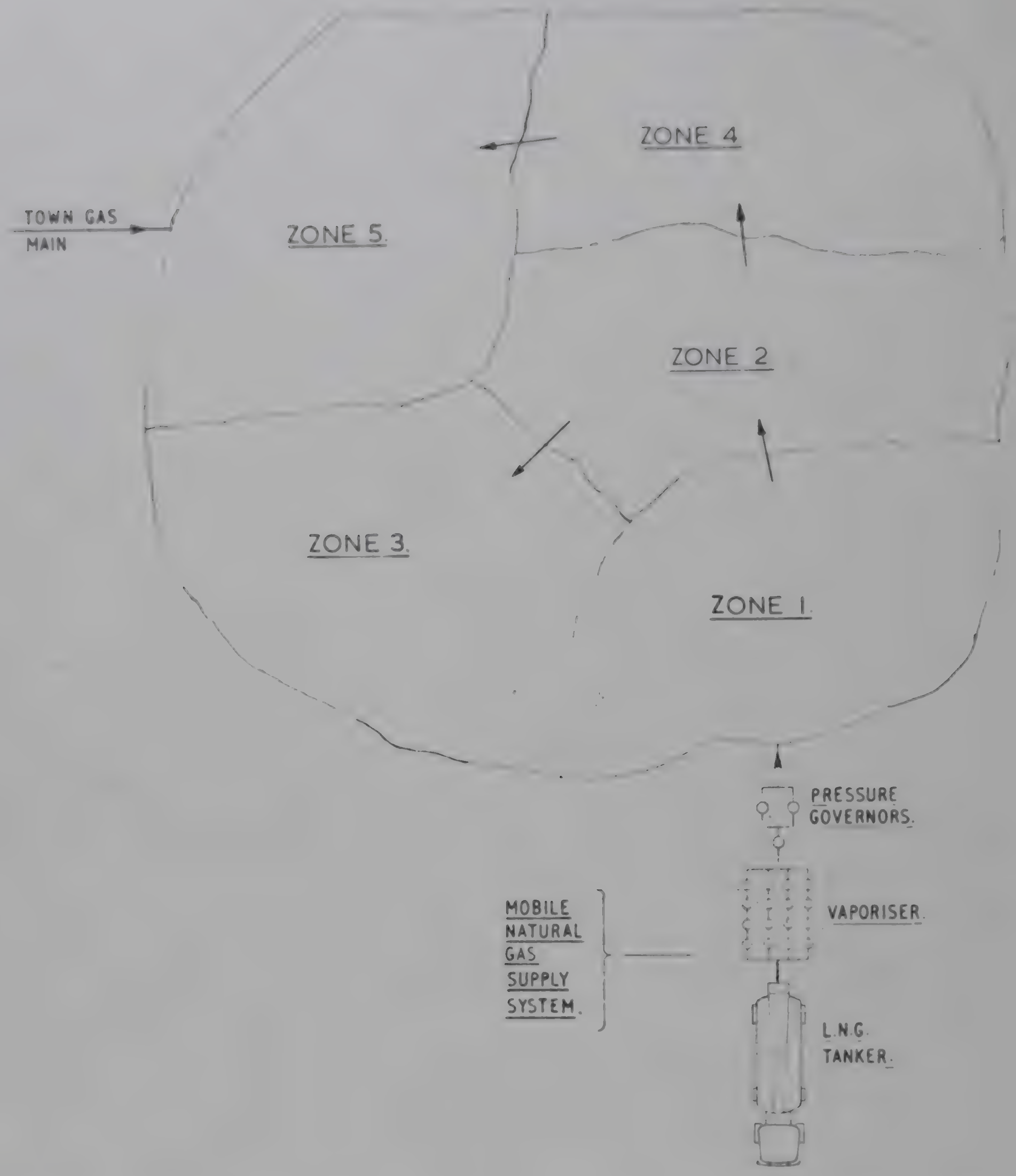

ZONE 4
TOWN GAS MAIN
ZONE 5.
ZONE 2
ZONE 3.
ZONE 1.
PRESSURE GOVERNORS.
MOBILE NATURAL GAS SUPPLY SYSTEM.
VAPORISER.
L.N.G. TANKER.

FIG. 5. TEMPORARY NATURAL GAS SUPPLY FOR SMALL TOWN DURING CONVERSION FROM TOWN GAS TO NATURAL GAS.

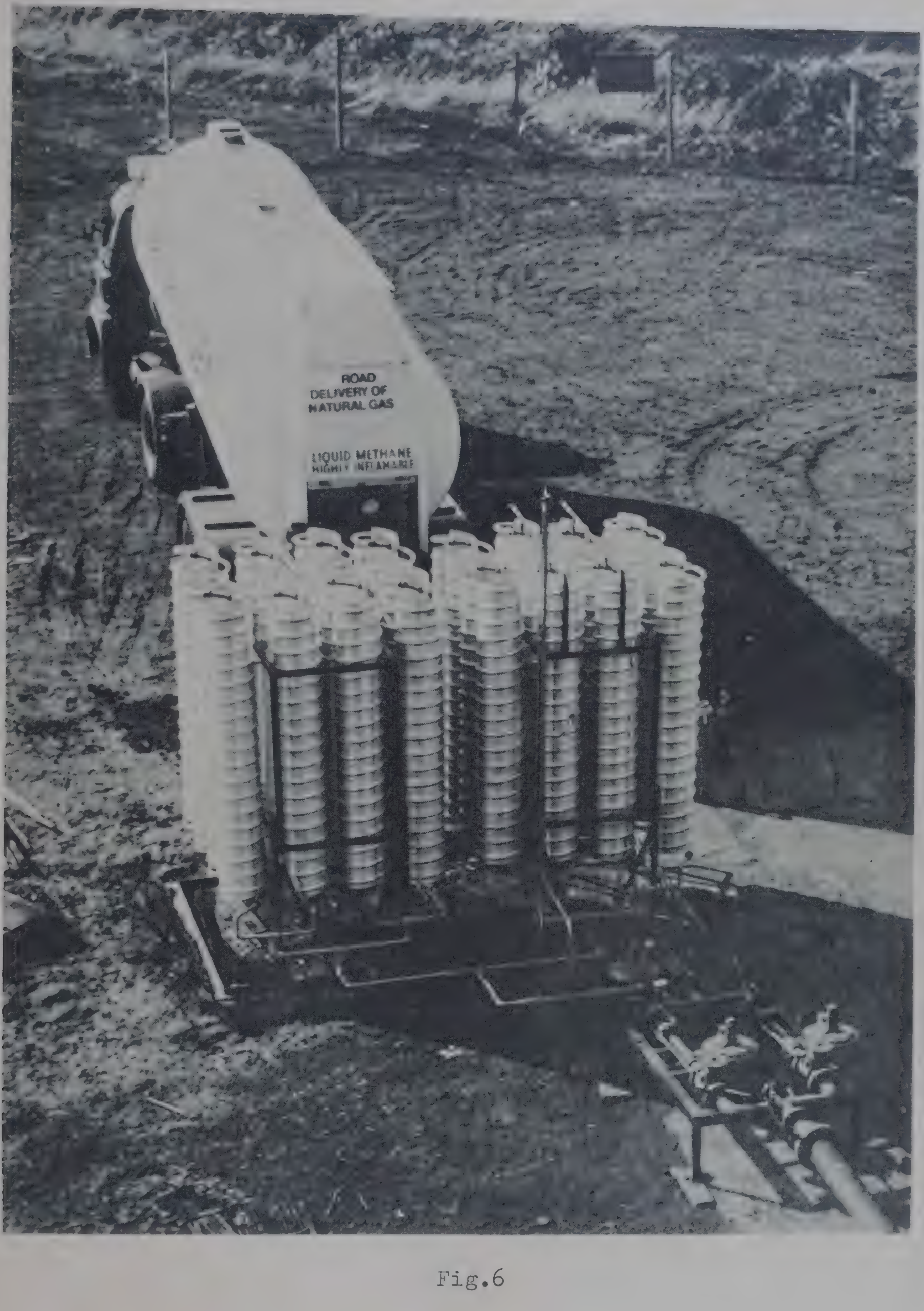
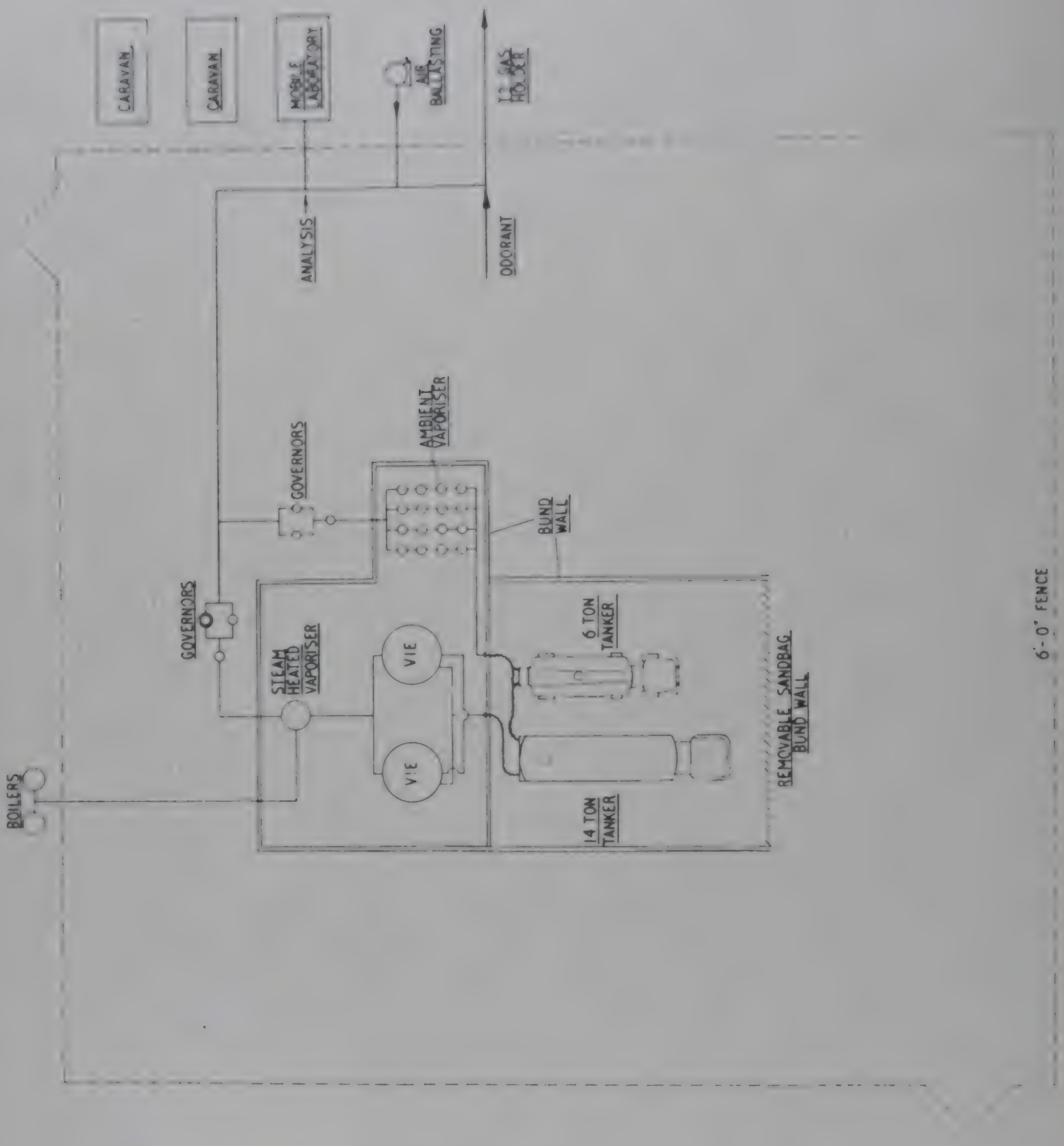

ROAD
DELIVERY OF
NATURAL GAS
LIQUID METHANE
HIGHLY INFLAMABLE

Fig.6


CARAVAN
CARAVAN
MOBILE LABORATORY
AIR BALLASTING
TO GAS HOLDER
ANALYSIS
ODORANT
GOVERNORS
AMBIENT VAPORISER
BUND WALL
GOVERNORS
STEAM HEATED VAPORISER
VIE
VIE
6 TON TANKER
REMOVABLE SANDBAG BUND WALL
14 TON TANKER
BOILERS
6'-0" FENCE

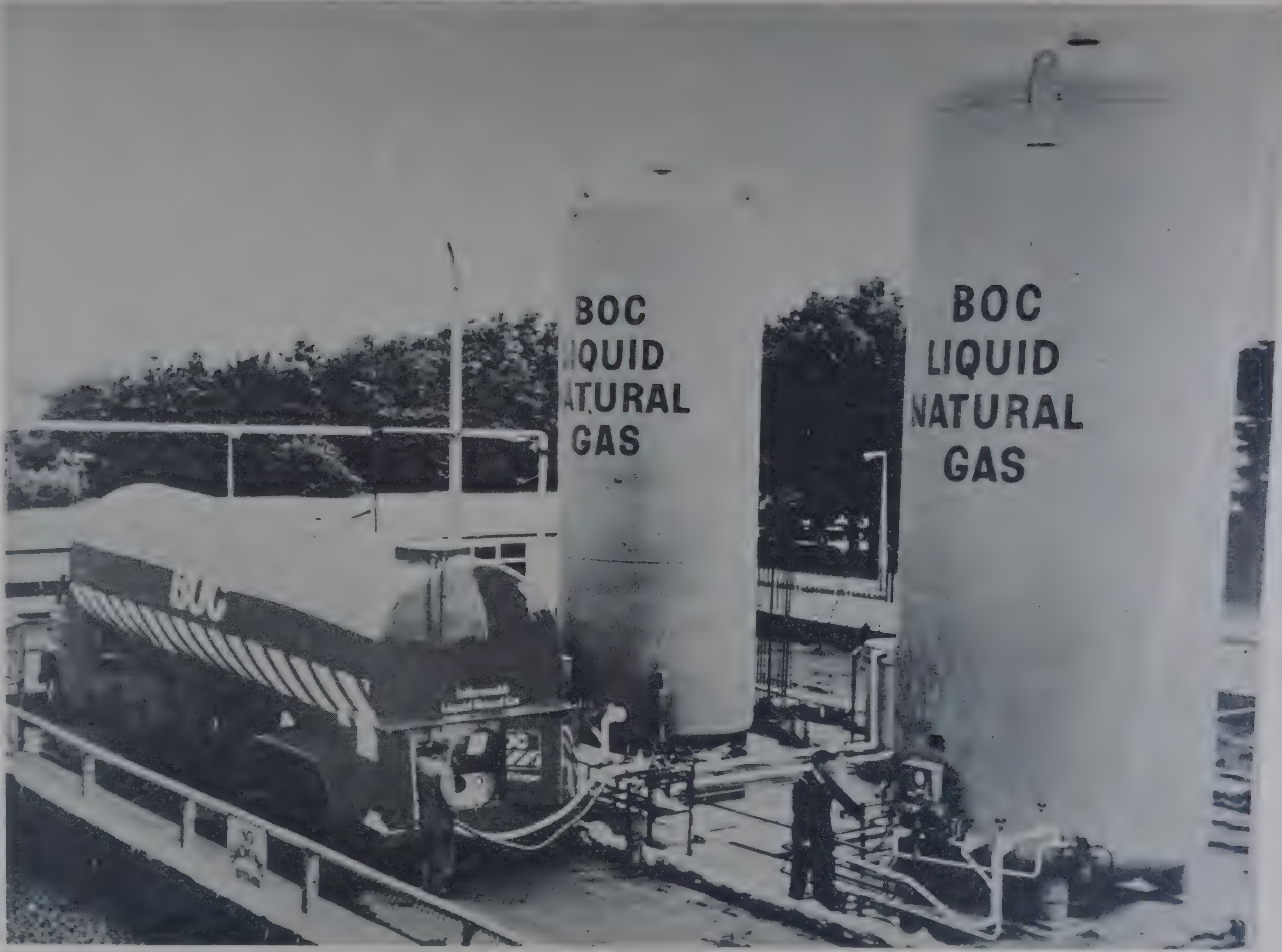

BOC
LIQUID
NATURAL
GAS
BOC
LIQUID
NATURAL
GAS
BOC

Fig.8

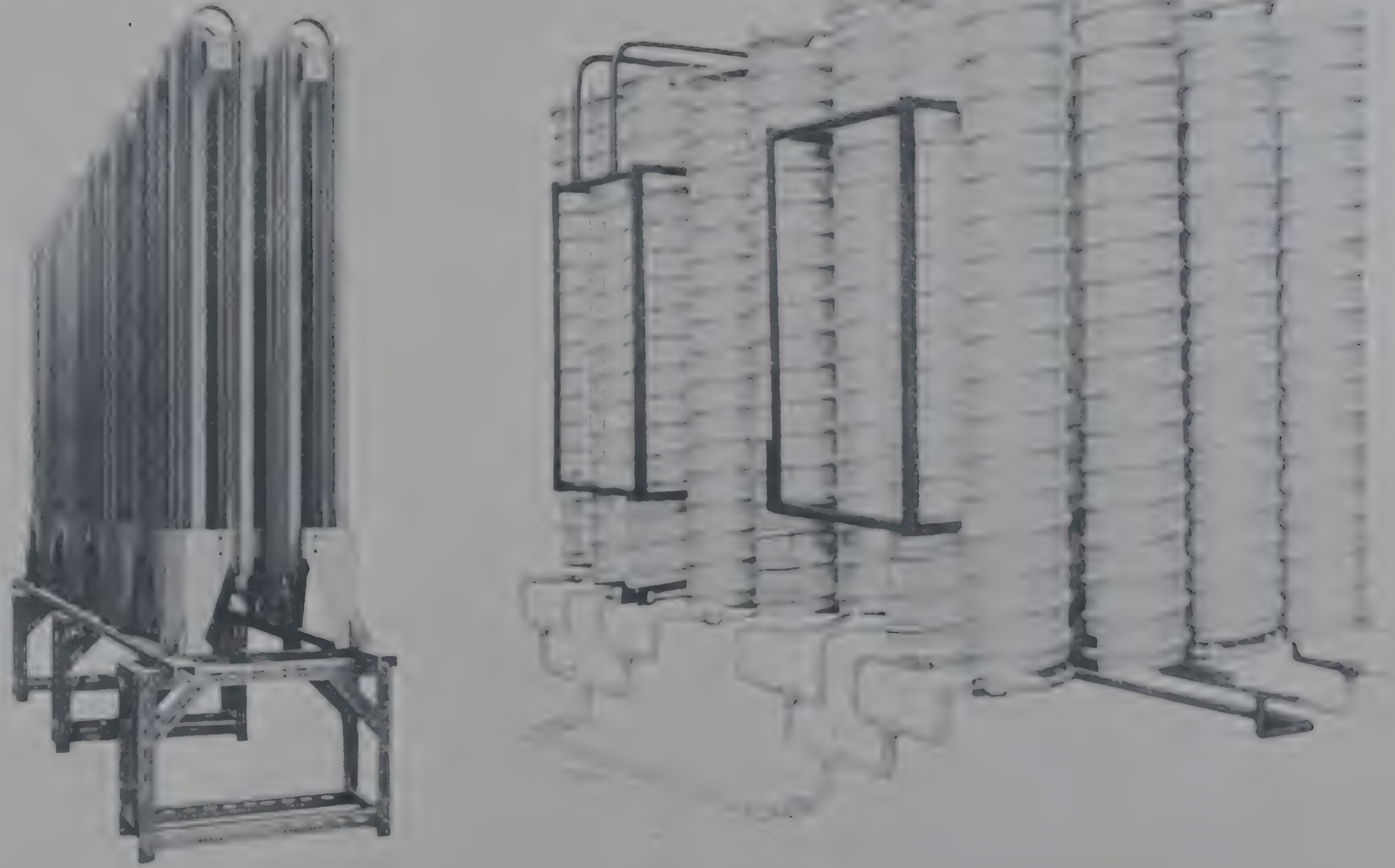


Fig.9

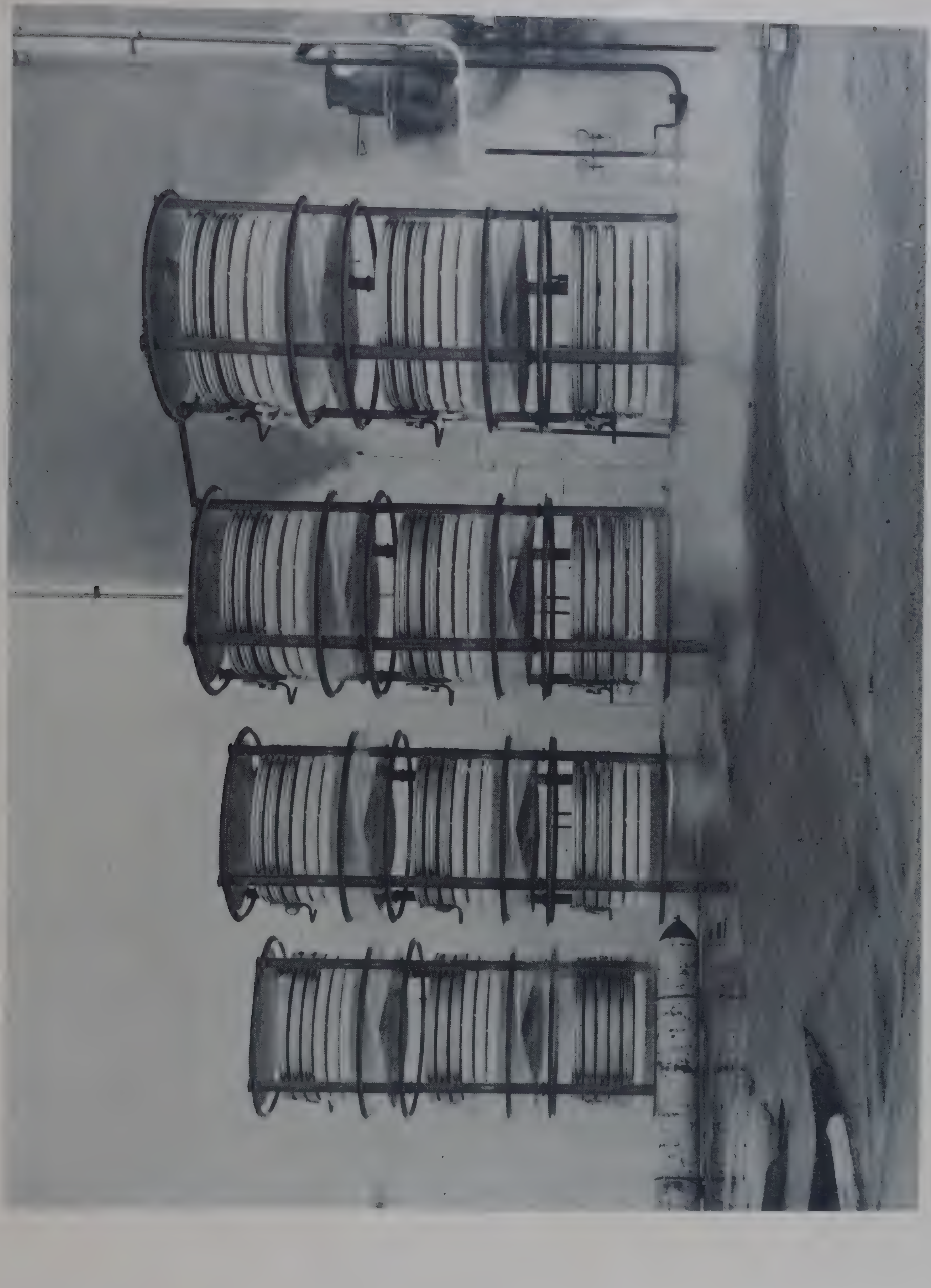


Fig.10

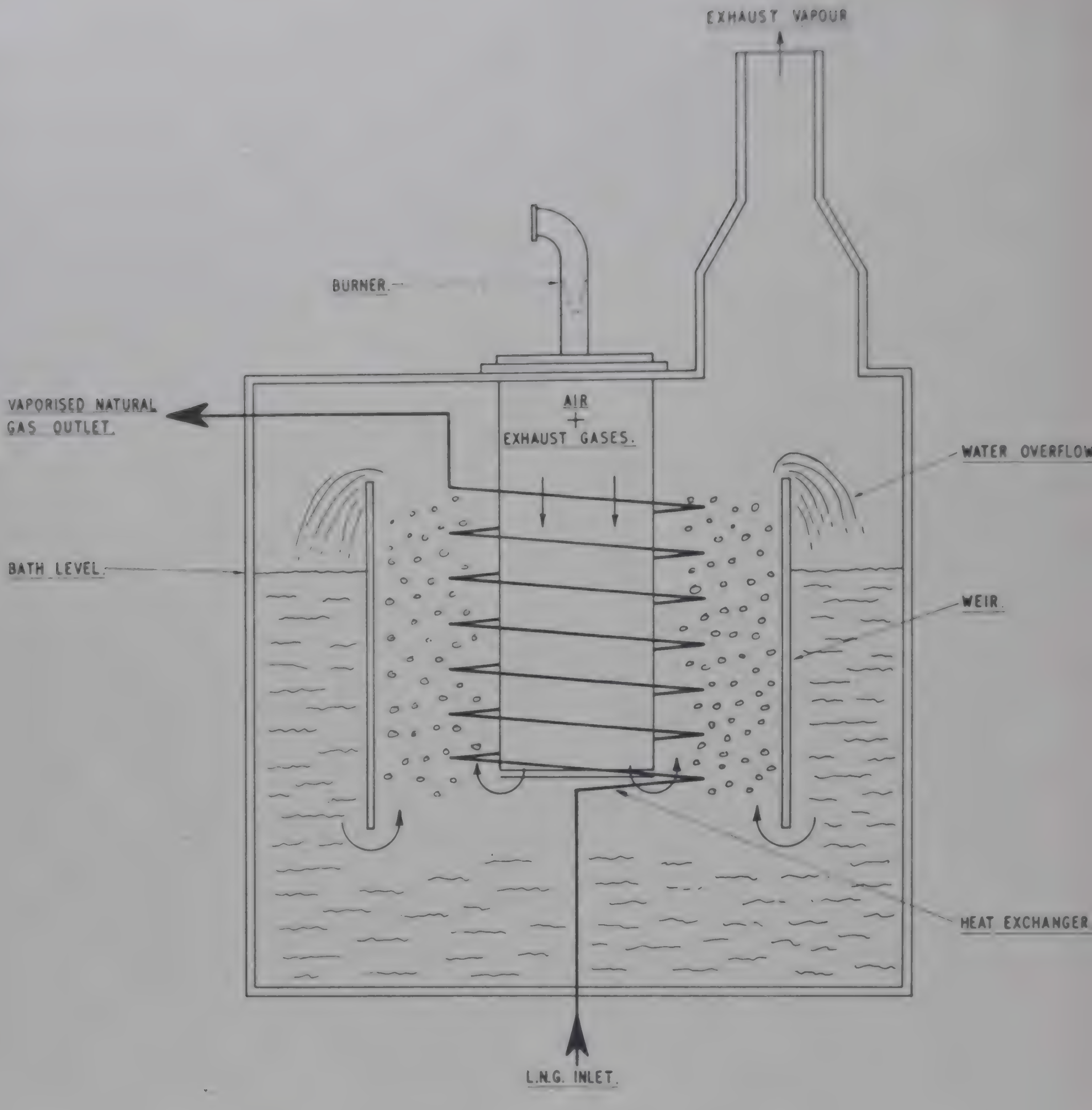

EXHAUST VAPOUR
BURNER.
VAPORISED NATURAL GAS OUTLET.
AIR + EXHAUST GASES.
WATER OVERFLOW
BATH LEVEL.
WEIR.
HEAT EXCHANGER
L.N.G. INLET.

FIG. 11. EXHAUST HEAT VAPORISER.

	%	B.P. °C
METHANE	86·6	-161·5
ETHANE	9·5	-103·7
PROPANE	2·7	- 42·1
ISO-BUTANE	0·4	-6·9
ISO-BUTANE	0·7	-0·5
PENTANE	0·1	36·1

FIG 12. COMPOSITION OF ALGERIAN NATURAL GAS.

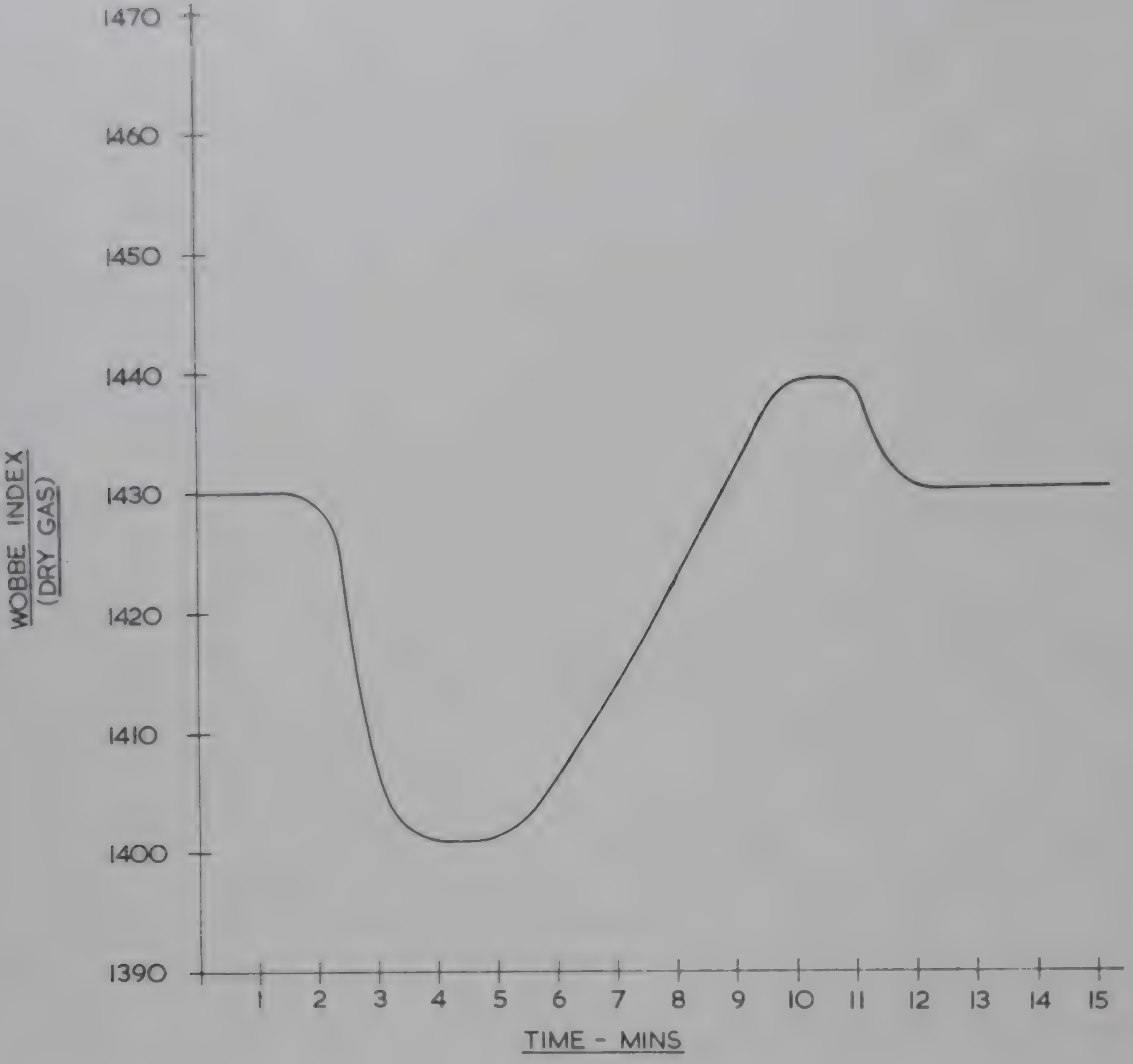


FIG. 13. VARIATION OF WOBBE INDEX (ALGERIAN GAS) ON REDUCING FLOW THROUGH AIR HEATER VAPORISER.

LNG FUEL/FREEZE SYSTEM

By J. T. Bernstein

INTRODUCTION

This decade has seen the commercial realization of natural gas liquefaction in the form of a number of plants built for peak-shaving and for gas transportation purposes; and this background, together with the projected growth of LNG production facilities makes it possible to consider applications of liquefied natural gas based on the prospect of LNG becoming a generally available and economically produced commodity in many areas, much as other liquefied cryogenic gases have passed from the stage of laboratory scale use to generally available commodities. One use of LNG which has been frequently discussed, and in a few cases tried on an experimental basis, is LNG as an automotive fuel. This paper discusses some general aspects of a specialized case of LNG as an automotive fuel: the use of LNG as both fuel and refrigerant in refrigerated transport vehicles. This rather limited application should have some added economic advantage relative to the general use of LNG as transport vehicle fuel, and could represent a logical initial area of introduction and testing.

LIQUID N_2 AND LNG FOR VEHICLE REFRIGERATION

In the early 1960's, the three major U.S. industrial gas companies started marketing liquid nitrogen refrigeration systems for transport vehicles as alternates to conventional mechanical (Freon) refrigeration systems. The main advantages of liquid nitrogen are elimination of mechanical refrigeration equipment maintenance, rapid cooldown or temperature recovery, and good temperature distribution and control. The major disadvantage appears to be cost. To some extent, the limited availability of liquid nitrogen may be a disadvantage in some areas, but on the whole, the achievement of widespread availability of liquid nitrogen has been a major basis for the success of the liquid nitrogen truck refrigeration scheme.

Based on the acceptance (still limited) of the liquid nitrogen transport refrigeration scheme, and on the beginnings of and future probability of widespread availability of liquid methane, it is suggested that a system using liquid methane for both fuel and refrigeration requirements in refrigerated tractor-trailers (and similar transport vehicles, and possibly fleet taxis) is worthy of serious study.

Some of the basic factors which appear to make such a system of interest are:

1. LNG costs should be competitive with liquid nitrogen cost, but LNG can also satisfy fuel needs. LNG may also be competitive (or nearly so) on a fuel basis.

2. For refrigerated trailers and continuous use, fleet vehicles (where frequent opening of the refrigerated space does not occur) fuel and refrigeration requirements for LNG appear compatible. Thus, substantial economies relative to the liquid nitrogen system appear possible.

3. LNG as a fuel has better characteristics with respect to:

    - Engine performance (higher octane, better burning efficiency)

    - Engine maintenance (elimination of combustion deposits, sludge)

    - Atmospheric contamination (substantial reduction of carbon monoxide and unburned hydrocarbons in exhaust)

The suggested scheme seems particularly timely in view of the current concern with air pollution, as well as the current development of LNG as an emerging commercial commodity. These factors, combined with the possibility of favorable economics, may make the proposed scheme seem attractive.

There are, however, many potential problems and negative aspects to the fuel/freeze scheme, including:

- Unbalance of fuel and refrigeration requirements
- LNG losses in idle time
- Safety aspects in LNG storage, handling, venting
- Government regulatory problems, including safety regulations and taxes
- Superior features of liquid nitrogen refrigeration in providing direct contact cooling and purging
- Engine conversion problems, problems in use of both LNG and conventional fuel, and connection between trailer (refrigeration) and tractor (fuel) systems in refrigerated tractor trailers.

None of these potential problems and disadvantages should prevent the proposed system from being attractive and successful, if a well engineered system with good economic advantage can be developed.

HISTORICAL BACKGROUND

The use of LPG (propane-butane mixtures) as automotive fuel has been frequently proposed, and often implemented since the early 1930's. A recent survey of refrigerated transport means by Levy(1) reports that in the USA in 1935, busses using liquid propane as both a refrigerant for air conditiong, and as fuel, were in use. Liquid methane has been used in Moscow as a truck fuel based on the installation of a peak shaving LNG plant in that city in 1948.(5) An article in a German refrigeration journal in 1963 proposed the use of LPG for both fuel and refrigeration in trucks for frozen food delivery.(6)

Significant experience exists in the use of both LPG and methane as internal combustion engine fuel. Almost all of the methane experience is based on large, stationary gas engines, and only a few automobile uses have been reported -- the use in Moscow of liquid methane, and the experimental vehicles being tested by San Diego Gas and Electric Co. The U.S. Steel Co. is currently using LNG as fuel for large ore carrier (off-the-road) transport vehicles.

COMPATIBILITY OF FUEL AND REFRIGERATION REQUIREMENTS

The quantity of liquid nitrogen required in vehicle refrigeration appears to be subject to considerable variation. In trailer units designed for use with liquid nitrogen refrigeration, superior insulation is recommended and liquid nitrogen vehicle refrigeration system manufacturers give an average annual consumption of 18 to 20 #/hr for frozen food transport in a 40 ft. trailer. This figure is based on geographical and yearly averaging of weather conditions. For non-frozen refrigerated trucks, average consumption may be 12 to 14 #/hr. A 19 #/hr nitrogen consumption is equivalent to 3000 BTU/hr, assuming the vaporized nitrogen leaves the truck at -60°F, and this value has been used as typical of nitrogen refrigeration systems.

A rather thorough N.B.S. program for testing cooling loads in commercial refrigerated trailers,(2) indicated an average of about 12,000 BTU/hr. Leo Block, in June 1966 ASHRAE Journal(3) gives 15,000 - 20,000 BTU/hr as a typical 40 ft. trailer cooling load, and 4¢/# as a typical liquid nitrogen cost, resulting in a $2.40/hr operating cost, which Mr. Block terms "considerably greater than mechanical refrigeration costs". Typical trailer mechanical refrigeration units are rated at 20,000 BTU/hr.

Some of the difference between 12,000 BTU/hr measured cooling load and the 3,000 BTU/hr equivalent to 19 #/hr liquid nitrogen is based on the difference between average ambient and 100° F ambient, but most of the difference seems to be based on the kind and degree of insulation. A typical description of a liquid nitrogen system states "most liquid nitrogen installations.... are in new vehicles (Polyurethane or styrofoam insulation) since relatively few older vehicles are sufficiently insulated for use with liquid nitrogen."

Assuming that average fuel consumption for a typical 40 ft. trailer-tractor unit will be equivalent (on a BTU basis) to diesel fuel consumption in the range of 100 to 200 B.H.P. output, liquid methane consumption for fuel use would be 32 to 64 #/hr.

Available refrigeration from this quantity of fuel consumed (counting latent head and sensible heat in warming to -80°F) would be between 10,000 and 20,000 BTU/hr. Thus, in steady running, liquid methane used for fuel in a refrigerated trailer would have a refrigeration capacity equal to that of typical mechanical refrigeration units, and substantially in excess of average requirement (3000 BTU/hr) in liquid nitrogen refrigeration applications.

For fleet taxi and similar applications, for air-conditioning automobiles in more-or-less continuous service, a liquid methane consumption of 10#/hr can be estimated for fuel use, which would supply 3500 BTU/hr (based on warming methane to 0°F). This should be adequate for taxi and similar automotive cooling applications. New York City bus air-conditioners are designed for about 100,000 BTU/hr air-conditioning load (because of frequent door openings) so this application (although of interest from the air pollution angle) does not seem feasible from the point of view of fuel-refrigeration load compatibility. Long distance buses should represent a compatible application.

BASIC ECONOMICS OF FUEL/FREEZE SYSTEM

The cost of liquid methane will be compared to costs for liquid nitrogen and to conventional fuel costs to establish a probable range of relative costs for LNG fuel/freeze vs. liquid N_2 and conventional fuel.

For a typical 40 ft. refrigerated trailer, the fuel and refrigerant costs, using conventional fuel and liquid nitrogen are about as follows: (U.S.A. costs)

Liquid Nitrogen: 19#/hr x 4¢/# = $.76/hr

Diesel Fuel: 5 to 10 gal/hr x 20¢/gal.*= $1.00/hr to $2.00/hr

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*Not including U.S.A. state and federal tax of about 10¢/gal.

Many estimates of the cost of liquefying natural gas have appeared based on the interest in LNG for transport economy and also for peak shaving. The peak shaving costs are apt to be high because of the very high storage requirements and non-continuous liquification unit operation.

Typical estimates of cost of liquifying methane which have been published are about 15¢ - 20¢/MSCF (or 0.4 to 0.5¢/#). This is comparable with the cost of nitrogen liquifaction: The "wholesale" large quantity contract price of liquid nitrogen is about $20/ton, or 1¢/#, of which (very roughly) 1/3 is gas purification and 2/3 liquifaction cost. As with liquid nitrogen for truck refrigeration, the actual use cost (3¢ - 5¢/# for N_2) would largely be associated with distribution costs rather than basic gas or liquifaction costs. As with nitrogen, liquid methane for the fuel/freeze application could be available from large scale production plants devoted mainly to other uses - peak shaving LNG and helium recovery plants, as well as LNG transport facilities. It has been estimated that in 1973 it will be possible to deliver LNG from Venezuela to the East Coast at a cost of about 45¢/MSCF, and typical estimates for large scale importation of LNG to the U.S. East Coast have been in the range of 45¢ to 65¢ per MSCF of delivered LNG. This is equivalent to a "wholesale" price of LNG at 1¢/# to 1.5¢/#. (For refrigeration use, liquid N_2 makes available only about 160BTU/#, as opposed to about 310 BTU/# for methane, so the .5¢ possible additional cost for LNG, on a "wholesale" basis is more than compensated.)

Thus, for refrigeration purposes, LNG should be quite competitive with liquid nitrogen: If LNG costs 4.5¢/# as opposed to 4¢/# for liquid nitrogen for truck refrigeration use, assuming a 0.5¢ basic price differential and equal distribution costs, liquid nitrogen refrigeration will cost 24¢/1000 BTU vs. 14.5¢/1000 BTU for LNG.

As vehicle fuel, the competitive pricing of LNG will depend heavily on distribution and handling costs which, as indicated above, are the major costs in the current relatively small scale liquid nitrogen use for truck refrigeration. Assuming gasoline or diesel fuel costs at about 20¢/gal. (not including taxes) on a retail basis, this is equivalent to 1.6¢/100,000 BTU. LNG would have to retail at 3.8¢/# to equal the same 1.6¢/100,000 BTU of fuel value. Apparently, LNG use as automotive fuel would have to involve a volume of use in excess of the present use of liquid nitrogen for vehicle refrigeration to reduce distribution costs to the point of being competitive. However, if even a relatively small part of the vehicle fuel market were assigned to LNG, this could provide enough volume to reduce distribution costs enough to make LNG competitive as an automotive fuel.

In analyzing the economics of the combined fuel/freeze use of LNG in refrigerated vehicles, based on the reported costs for liquid nitrogen (3-5¢/#, 4¢/# average in truck refrigeration use) a price close to the competitive fuel price should be feasible without the economies of larger scale use of LNG. Assuming the 0.5¢/# differential mentioned over the 4¢/# N_2 average, costs for the LNG fuel/freeze system should be competitive -- on the same basis as the typical liquid nitrogen - diesel fuel truck costs given above, as shown in Table I below:

Table I

Comparison of Liquid N_2 and Diesel Fuel Cost vs. Fuel/Freeze LNG

Liquid N_2 and Diesel Fuel

Avg. Fuel Equiv. power use	100 BHP	200 BHP
Gal/hr Diesel Fuel	5	10
Hourly fuel cost @ 20¢/gal.	$1.00	$2.00
Liq. N_2 cost @ 4¢/# for 3000 BTU/hr	.76	.76
Total Cost/hr (ref'n. 1/2 time)	$1.38	$2.38
Total Cost/hr. (continuous ref'n.)	$1.76	$2.76

LNG "Fuel/Freeze" System

Avg. Fuel Equiv. power use	100 BHP	200 BHP
#/hr LNG for fuel	32	64
Ref'n. available from LNG, BTU/hr	10,000	20,000
LNG cost @ 4.5¢/#	$1.44	$2.88

Consideration of more basic aspects of raw material and production cost can lead one to think that LNG can, if used in sufficient quantities, be a cheaper automotive fuel (for fairly continuous operation) than petroleum products: Assuming the cost of gasoline at a refinery is 10¢/gal. - 1.63¢/# = 8.0¢/100,000 BTU; LNG cost of 60¢/MSCF = 6.0¢/100,000 BTU.

The 60¢/MSCF production cost would be achievable in large scale production under present conditions. If the projected 45¢/MSCF cost for foreign LNG delivered to the U.S. can be achieved, LNG would represent a high octane, nearly ideal motor fuel available at nearly half the production cost of present petroleum fuels.

Consideration of some of these basic aspects (including refrigeration aspects of LNG) have lead to the projection of 30% lower operating costs for supersonic jet transport using LNG fuel instead of kerosine type jet fuel.(4) This projection was based on 1.5¢/# or 60¢/MSCF cost for LNG.

Thus, conclusions which can be drawn from this approximate, "broad-brush" treatment of LNG economics in automotive use are:

- LNG may be of interest and possibly competitive for small-scale use in refrigerated transport vehicles when compared to the use of liquid nitrogen and conventional fuel.

- LNG could be of interest in continuous use vehicles of fuel if widespread use reduced distribution costs.

MARKET DEVELOPMENT

Based on the above technical and economic considerations, the following overall market development suggests itself for an organization interested in promoting LNG as an automotive fuel:

1. LNG can initially be introduced (at a price of about 4¢ to 4.5¢/#) in a system to provide both fuel and refrigeration for trucks, busses, fleet taxis. Customers with existing liquid nitrogen truck refrigeration equipment should be particularly interested, since they already have cryogenic storage equipment which might be adapted to LNG at minimum initial investment cost. An LNG truck system could handle economically significantly larger refrigeration loads (up to 20,000 BTU/hr) than are presently economical with liquid nitrogen (2,000 - 4,000 BTU/hr).

2. Based on development of an acceptable LNG fuel/freeze system, establishment of a distribution system, recognition of the advantages of methane as a fuel, and lowering of costs through increased use, LNG could become utilized in many automotive applications not involving refrigeration requirement, but involving continuous fleet type use - trucks, busses, taxis.

3. The development and establishment of know-how, techniques, personnel, experience, systems and components, as well as production and distribution facilities for LNG automotive fuel could enhance the possibilities of developing the potential SST jet plane market for LNG.

NEGATIVE ASPECTS AND SYSTEM DESIGN PROBLEMS

The proposed "fuel/freeze" system, combining the use of LNG for fuel and refrigeration purposes, has many negative aspects and design problems which should not be overlooked or underestimated.

In the economic area, no allowance has been made for LNG loss through boil-off during vehicle idle periods. Although this would not be significant in continuous use vehicles, it obviously becomes a problem as vehicle use departs from the continuous ideal. In addition, any overall economic evaluation must include equipment as well as operating costs. The extent to which equipment costs for the LNG system are greater (or less) than alternate system cost may have a significant effect on economics.

The actual design of the proposed LNG fuel/freeze system presents many problems. Some of the problem areas are listed below:

1. Allowing for unequal loads for LNG for fuel and refrigeration.
    - Should vapor storage be provided when more gas is vaporized for refrigeration than for fuel?
    - Should excess refrigeration vapor be vented?
    - Quantity of storage, number of tanks?
2. Connection of tractor and trailer units.
    - Should each have separate tanks?
    - Should alternate fuel storage be provided for tractors? Should LPG or conventional fuel be used as "bank up"?
    - Reliable interconnecting devices and system.
3. Safety problems:
    - How to dispose of vented gas.
    - How to eliminate uncontrolled spills, leaks, losses of LNG.
    - I.C.C. and other safety regulations.

    Possible solutions: Catalytic burning of vented gas, pressurized gas storage. Development of safety standards and regulations in peak shaving and LNG shipping should help define safety approach for fuel/freeze system.
4. Overall fuel system and engine modification:
    - Should trucks be adapted for LNG only, for for dual fuel use?
    - Should LNG be removed from trucks and returned to central storage when trucks not in use? How to handle general problem of LNG vaporization losses?

- To what extent should engines be modified to take advantage of potentially better performance on LNG?

5. Refrigeration system design:

    - Since LNG refrigeration will be indirect (as opposed to direct spraying of liquid N_2 into refrigerated areas), problems of circulation and temperature control are different. Probable approach to this would be similar to freon unit circulation/control scheme.

Finally, there are overall problems of social utility involved in considering the specific LNG fuel/freeze scheme, or the general use of LNG as automotive fuel; such as safety aspects, relation to air pollution, etc.

Acknowledgement

Most of this paper is based on a study performed for Cities Service Oil Co. in June 1967, and the permission of Cities Service to publish this study is gratefully acknowledged.

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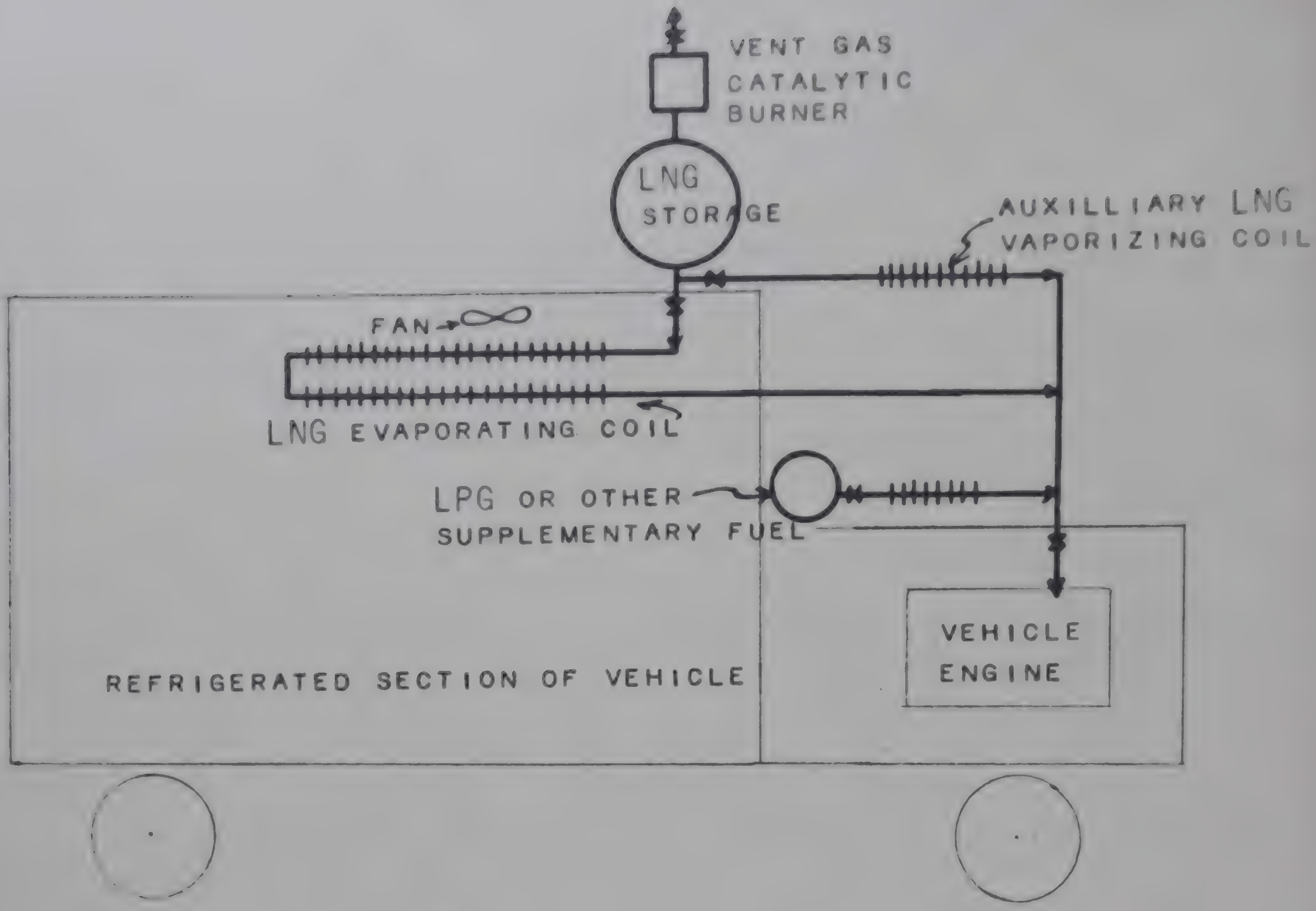
VENT GAS
CATALYTIC BURNER
LNG STORAGE
AUXILLIARY LNG VAPORIZING COIL
FAN
LNG EVAPORATING COIL
LPG OR OTHER SUPPLEMENTARY FUEL
VEHICLE ENGINE
REFRIGERATED SECTION OF VEHICLE

Figure 1. LNG Fuel/freeze system - simplified diagram

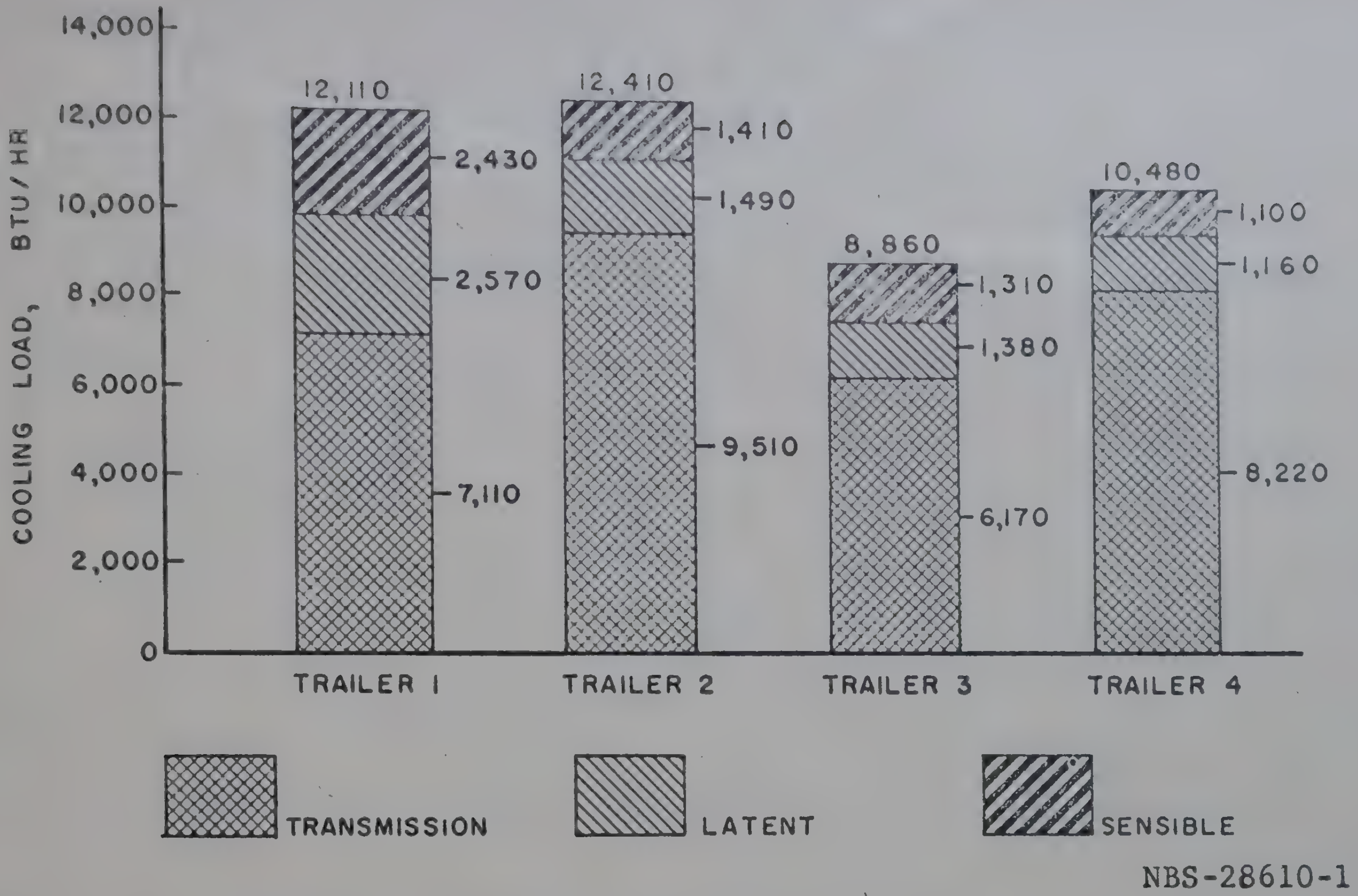

Figure 2 -- Computed total and incremental cooling loads of four commercial trailers at a road speed of 50 m.p.h. extrapolated to ambient temperature of 100° F. and relative humidity of 50 percent. Cargo space temperature 0° F.

(FROM NBS MARKETING RESEARCH REPORT NO. 433 - REFERENCE 2)

THE DESIGN OF OPTIMUM LNG HIGHWAY TANKERS

By W. N. Latham*

ABSTRACT

With the recent surge of interest in and availability of LNG, a demand has arisen for optimum design LNG highway transport tankers. This paper reviews design criterion of cryogenic highway semi-trailers in general and of LNG trailers in particular. Special emphasis is given to methods of trailer unloading and to safety. Design and performance data are presented on three 11,650-gallon trailers designed and built specifically for LNG service by Process Engineering Inc.

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*Process Engineering Inc.

INTRODUCTION

Over the highway movement of LNG has been underway on a small scale for several years. Experience gained has been primarily with used equipment originally designed for liquid nitrogen service. Nevertheless, these operations have been quite successful and have demonstrated the basic feasibility of existing cryogenic tanker technology to LNG service. It is thus logical that the next step is the design and fabrication from the ground up of a new cryogenic LNG trailer. This step has now been made and will be reported on at the end of this paper.

It is first proposed however, to cover in detail the major design alternatives considered in arriving at a cryogenic trailer design. Although there is much in common between the design of an optimum nitrogen trailer and an optimum LNG trailer, there are also some very basic differences. Representing a company involved in the design and manufacture of cryogenic trailers, the author intends in this paper to cover these similarities and differences in detail for the benefit of those who are not versed in cryogenic trailer design and operation. It is hoped that this information will be helpful to anyone interested in LNG systems involving the use of highway tankers, whether his concern be long range planning or detail design.

TRAILER DESIGN GOALS

The major objectives of a cryogenic trailer design are easily listed, but, unfortunately, are not easily separated from each other. Design practices which effect one design goal favorably often effect others adversely. Before embarking on the detailed discussion of each, the following list of major objectives with brief comments is presented.

Weight Optimization: Because the operating costs of a tractor-trailer combination, as represented by driver wages and benefits, tractor fuel and maintenance, and insurance and taxes are high and are largely independent of the amount of payload moved, the trucking industry is continuously striving to increase the payload capacity of its equipment. Since gross vehicle weights are set by law, the technical problem for most liquid haulers becomes one of minimizing the trailer tare weight and maximizing its gross capacity. Because of its low density, LNG adds to the above the problem of staying within legal dimension limits.

System Compatibility: As new and different applications of LNG trailers present themselves, it becomes apparent that there is no single transport design which is best suited for every application. Wide variation may exist, for example, as to the most suitable method for trailer unloading. An unloading system which is well suited to one operation may be grossly inadequate in another. The trailer loading method must also be considered as well as such factors as, for example, the permissable heat leak to the contents.

Operating Reliability: A certain amount of unplanned maintenance and down time is expected by all trailer operators. Companies with large fleets maintain well trained and stocked service departments and are usually able to cover with other equipment when a trailer is out of service. The small fleet operator, such as most LNG haulers will be at first, has a more serious problem. Since he can afford only the barest minimum of back-up equipment, he must choose trailers with the greatest built-in dependability, even if it means, for example, paying a higher initial cost or sacrificing operating convenience.

Safety: Both the designer and user of LNG transport equipment bear a heavy responsibility to the general public and to the gas industry to see to it that every reasonable safety precaution is taken. In addition to the standard safety devices employed on conventional cryogenic trailers, a number of others are recommended for LNG service.

WEIGHT LIMITED VS. VOLUME LIMITED DESIGNS

Despite the basic similarities between trailers designed for LNG and for liquids oxygen, nitrogen, and argon, there is one difference in fluid properties which has major design ramifications. This difference is in the low density of LNG and its relation to highway dimension and weight limitations. In the U.S.A. highway laws limit semi-trailers in most states to overall lengths of 40 feet (12.2 M) and overall widths of 8 feet (2.44 M). In addition, gross weights including the tractor are limited generally to 73,280 pounds (33,300 K.G.). Trailers which are designed for hauling liquid oxygen, nitrogen and argon are weight limited. In other words, gross weight limits are reached with trailers of less than maximum permissable dimensions. Trailers for use with LNG, on the other hand, are slightly volume limited. Full legal width and length are met somewhat before maximum gross weight limits are reached.

This factor throws much of the design philosophy and thinking of the traditional oxygen, nitrogen or argon trailer into a new light. Whereas previously every effort was made to reduce the tare weight of the trailer in order to maximize the payload, often at the expense of operating flexibility and reliability, it becomes possible in LNG service to include previously eliminated design features without sacrificing payload.

UNLOADING METHODS LISTED

Probably the second most important consideration in cryogenic trailer design and closely related to the problem of weight optimization is the unloading method. Depending on the particular application for which the trailer is intended any one or combination of several unloading methods is preferred. The particular unloading method chosen is likely to have important implications throughout the design through its dictation of, for example, the inner vessel design pressure, and both internal and external piping.

A listing of various unloading methods follows with a brief description of each. After that the advantages and drawbacks of each under different operating conditions are discussed.

Trailer mounted pumps consist generally of an electrically driven centrifugal pump permanently mounted and piped on the trailer. The power source is either from a power-take-off driven alternator on the tractor or from an electric outlet at each pumping location. A few trailers have been outfitted with alternators driven by auxiliary gasoline engines. A few also have been outfitted with straight mechanical pump drive systems. These consist of a line shaft from the tractor power-take-off running the length of the trailer to a V-belt pump drive.

Trailer mounted submerged pumps are a variation of the above arrangement and consist of an electrically driven centrifugal pump mounted inside of the cargo vessel, submerged in the liquid payload. Electric power leads are brought out through a hermetic seal to an external power supply. Usually a manway through the vacuum space is provided for access to the pump.

Ground mounted pumps are electrically driven centrifugal pumps located at and piped permanently to a storage tank, thus eliminating the need for a trailer mounted pump.

It should be pointed out that for all of the above pump unloading systems the trailer tank need have only a high enough pressure rating to supply suction head to the pump. A pressure of 25 psig (1.8 kg/cm^2) is generally adequate.

Pressurized tank transfers are accomplished by designing and fabricating the trailer tank for working pressures slightly greater than the intended receiver pressure and pushing the liquid across with gas pressure. The pressure in the trailer is maintained either from an external supply of pressurized gas, or more commonly by evaporating a portion of the liquid payload in an external ambient heat exchanger, commonly referred to as a pressure building coil, and piping it back into the top of the tank.

UNLOADING SYSTEMS COMPARED

Due to the nature of their business and to the fact that their trailers are weight limited, the industrial gas industry has widely adopted electrically driven trailer mounted pump unloading systems for liquid oxygen, nitrogen and argon deliveries. In general, each of their trailers serves many storage tanks, making a ground mounted pump at each storage tank substantially more expensive than a single pump for each trailer. Also, since their storage tanks often range in pressure up to as high as 300 psi, to design a trailer for pressure transfer would result in substantial loss of carrying capacity due to the increased weight of the high pressure trailer tank. Thus they have settled on trailer mounted pumping systems as the best system.

Inherent problem areas with such an unloading system center around maintenance and operating reliability and safety. These systems have always been faced with the necessity for frequent pump shaft seal replacement, and with electrical power supply and control problems. Pump shaft seal leakage not only results in down time and repair expense, but also, in LNG service, in a fire hazard. All of the previously listed trailer pump drive systems, due to the severe road conditions encountered of dirt, moisture, and vibration, have historically had poor reliability and high maintenance costs. The consequences of these problems have been alleviated to a certain extent for the industrial gas people by the fact that they have large fleets with a fair amount of old equipment available for back-up service.

A secondary problem area with pump unloading systems is the complexity of the associated external piping and the increased likelyhood of operator error.

Submerged pumps are an interesting variation to the trailer mounted pump concept. These have the advantage that pump shaft seal requirements are no longer critical. With them it is unimportant if a portion of the pump throughput leaks through the shaft seal and back into the tank. Submerged pumps are designed with minimal shaft sealing and thus eliminate the chief maintenance and safety problems of external pumps. In addition, due to improved motor cooling and reduced electrical resistivity of the windings, the performance of a submerged motor is improved, permitting the use of a smaller motor to produce the required pump horsepower. Another operating advantage of the submerged pump is that since it is inside the cargo tank and always cold, liquid losses associated with the cooldown of and heat leak to an externally mounted pump are eliminated. This advantage,of course, is offset by the fact that heat generated by the motor goes directly into the liquid rather than to the atmosphere and that if a manway through the vacuum space is provided for access to the pump, an additional steady heat leak to the payload results.

The other primary disadvantage of submerged trailer pumps is the added initial cost to cover installing the pump, wiring the motor, and providing a means of access through the vacuum jacket for maintenance when required.

Ground mounted pumping systems have the advantage over trailer mounted pumping systems of reducing the tractor and trailer weights by an amount equal to the weight of the P.T.O.,alternator, pump drive motor, and pump and associated piping. Ground mounted pumps are less expensive per pump than trailer mounted pumps, and since they are not continuously bouncing along the highway, generally require less maintenance. Thus for bulk hauling operations with weight limited trailers in which one or more trailers are delivering liquid to a single site, ground mounted pump unloading is the logical choice. The disadvantage of ground mounted pump unloading, of course, is the fact that the trailers are quite inflexible and can unload only where there is a ground mounted pump or the receiver pressure is less than the trailer working pressure.

Pressure unloading systems have the advantage of simplicity of piping and operation and highest reliability. With the simplest of all piping systems and with no moving parts other than valves, practically all possibility for human error and for mechanical breakdown is eliminated.

On weight limited trailers, however, the extra weight of the inner vessel designed for full unloading pressure rather than just pump priming pressure is apt to result in increased initial cost and loss of carrying capacity as compared to a pump unloading system. This is particularly true for applications requiring unloading pressures of greater than about 60 - 100 psi (4.0-7.0 kg/cm^2), as it is in this range that the added weight and added initial cost of a pressure transfer unit exceed the weight and cost savings resulting from the elimination of the pumping system.

In addition to the above considerations, thought must be given when analyzing a pressure transfer system to the source of the pressurizing gas and to its disposal once the transfer is completed. This gas is normally supplied by evaporating part of the liquid payload, which of course means that part of the liquid load instead of being delivered as a liquid is left in the trailer at the end of the transfer in the form of a compressed gas. The economic loss due to this liquid having been converted to gas thus becomes an operating expense. This expense can be reduced, if means is available either at the unloading location or back at the loading plant to recover or reliquefy part or all of this gas. This loss is partially compensated for in comparison to a pump unloading system by the reduced piping cooldown liquid losses at the start of the transfer.

Typical piping diagrams for pressure transfer and trailer mounted pump transfer LNG trailers are shown in figures 1 and 2. As would be expected, the pressure building circuits and gas phase side piping, including the main tank safety valve and bursting disc are very similar for the two. The main difference lies in the extra valves on the pumper unit in the liquid line and in the pump suction, pump cooldown, and pump discharge circuits.

SAFETY CONSIDERATIONS

Pressure vessel safety valves are shown on the piping schematics and include the main tank pop safety and bursting disc safety, and pop "line safety" valves wherever liquid may become trapped between two valves. The pressure transfer unit also has a low pressure, usually 15 - 25 psig (1.0 - 1.8 kg/cm^2), road safety for low pressure tank relief while on the highway plus an isolation valve to permit building the tank to full pressure for unloading. A low pressure safety is recommended and in some cases required by law for over the road non-vented operation. The heat leak into the inner tank raises the temperature and saturation pressure of the liquid. Since the tank is in motion and the liquid is thoroughly mixed, stratification does not occur as with a stationary tank. As the liquid is warmed it expands, filling the gas space left above the liquid at the time of loading. The low pressure relief valve must be set to relieve before the expanding liquid covers the vent line opening. If this did not occur, the vent would blow liquid instead of gas when the safety finally did relieve. The setting of the low pressure safety is thus a function of the intended original filling level, the higher the level, the lower the required safety setting.

In addition to the above safety valves, which are pretty much standard on all cryogenic trailers, other safety items should be considered because of the extra hazard associated with LNG. These include a swing check in the pressurizing coil return line to prevent escape of gas from the trailer vapor space back through a leak in the coil resulting, for example, from a highway accident. Another desirable safety feature is some type of emergency shut-off valve in each of the liquid lines. One arrangement is for these to be quick closing valves equipped with springs to close them and held open by a cable and fusible link to the front of the trailer. In the event of a fire the fusible link will melt, releasing the valves to be closed by the springs. Also, if for some reason the driver did not have access to the manual valves, by releasing the cable at the front of the trailer these valves would be closed by the springs.

All overboard venting, with the exception of the trycocks, should be done upwards and high off the ground. The thinking here is that the gas vapors will tend to rise and should be vented as far away as possible from possible sources of ignition and where they will rapidly disperse with the surrounding air.

In addition to the above safety features, which are specifically related to the payload, two commercially available trailer safety devices are of interest. The first is aimed at eliminating tire fires, which are a fairly common occurence with dual tired vehicles and constitute a particular hazard on an LNG trailer. Tire fires usually result fron one of a pair of dual tires being run flat or at reduced pressure, unbeknown to the driver. The Scovill Schrader Company supplies a system which automatically bleads air from the trailer air brake system and feeds it to the tires, providing makeup for leaks,maintaining them at a preset pressure. If a serious tire leak develops which the system cannot keep up with, a warning light is turned on.

The second item is the Hope Stabilizer, which is a rotating king pin device, brake loaded at the time of trailer brake application to reduce the likelyhood or trailer jacknifing. This equipment is readily installed on the trailer in place of the standard kingpin and is an attractive safety extra.

PROCESS ENGINEERING LNG TRAILER

As stated earlier, Process Engineering Inc. last fall delivered three trailers designed and fabricated specifically for LNG service. These units will be used primarily for hauling liquid to satellite storage tanks from a central liquefication plant. A pressure transfer design was chosen for its inherent reliability and simplicity of operation. Figure 3 is a photograph of one of these units. It has a gross capacity of 11,650-gallons (44,100 liters) and a tare weight of 19,400 pounds (8,800 kg.). With a working pressure of 56 psig(3.9 kg/cm^2), it is intended for pressure transfer unloading into receivers of up to 50 psig (3.5 kg/cm^2) working pressure or ground mounted pump unloading into receivers of higher pressure. The inner vessel and outer vacuum jacket are both fabricated of 5083 aluminum. The tandem suspension is a Hendrickson model RUEA-340 extended leaf spring unit chosen for its excellent handling and full and empty ride characteristics. Rockwell Standard 4" x 6" rectangular axles with cam type air brakes and spring loaded parking chambers are used.

Figure 4 is a photograph of the control piping. The flow schematic is the same as shown on figure 1. Loading and unloading is through the main liquid line and is controlled by the "main liquid" valve. Pressure is increased by opening the "pressure build" valve and decreased by opening the "blowdown" valve. Top filling and/or gas recovery is through the auxiliary top line controlled by the "gas return" valve.

At the time this goes to press these trailers have just gone into service, having hauled between them on their maiden trips approximately 6,000 ton miles of LNG. Unloading has been by pressure transfer into a low pressure receiver. Unloading times for full loads of liquid have been less than one half hour. The boil-off due to heat leak has been measured at 0.7% per day of the contents. This heat leak results in a non-vented rate of pressure rise on the highway of approximately 2.5 psi (0.18 kg/cm^2) per day.

In conclusion, the underlying objective of this paper has been to show that LNG highway tankage is not something of the future but rather of the here and now. Thanks to the requirements of the industrial gas industry over the past fifteen years, the technology of cryogenic trailer design and fabrication has reached a state of high maturity and sophistication. A real capability in these areas exists and is ready and able to meet the requirements of the LNG industry.

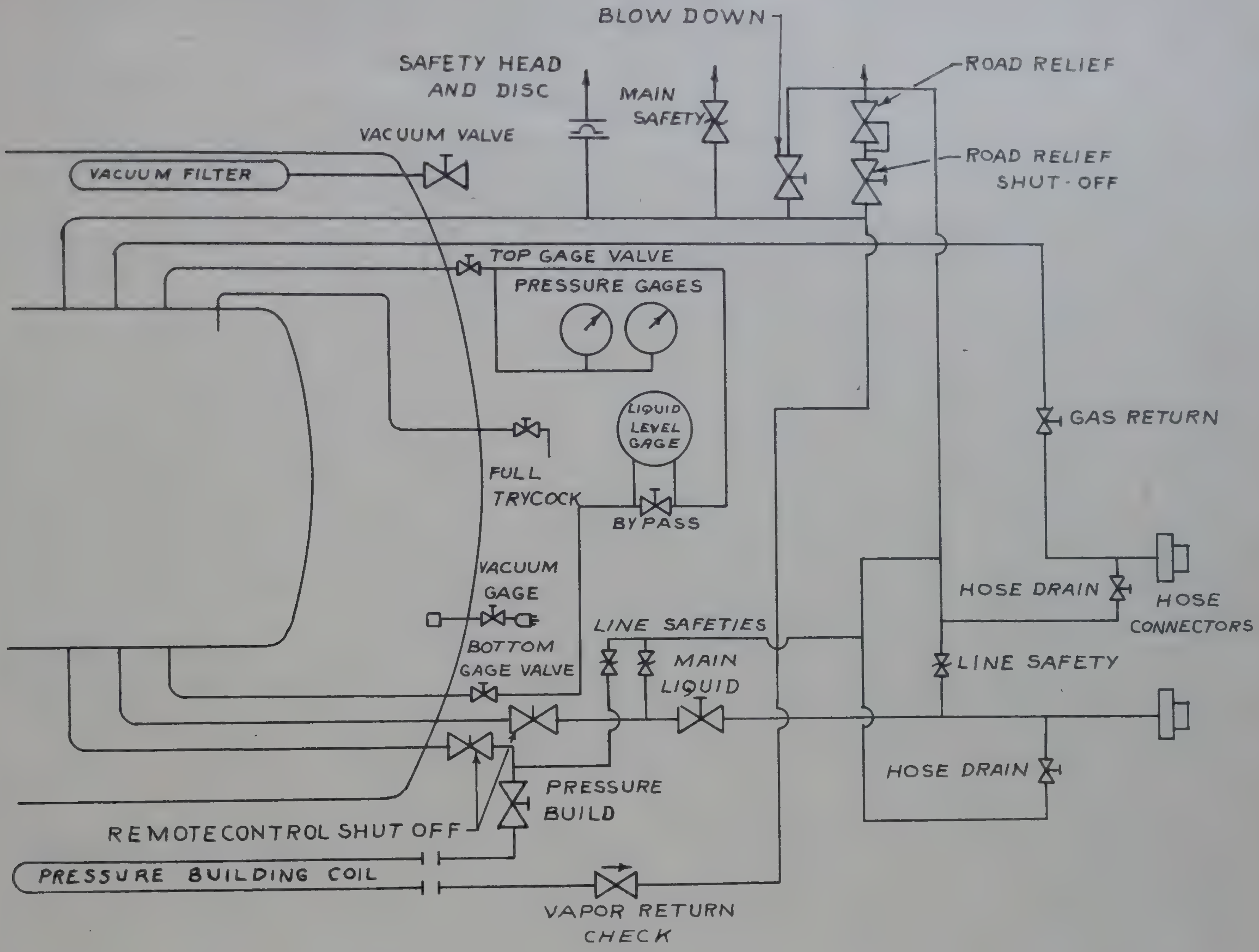
BLOW DOWN
SAFETY HEAD AND DISC
MAIN SAFETY
ROAD RELIEF
VACUUM VALVE
VACUUM FILTER
ROAD RELIEF SHUT-OFF
TOP GAGE VALVE
PRESSURE GAGES
GAS RETURN
LIQUID LEVEL GAGE
FULL TRYCOCK
BY PASS
VACUUM GAGE
HOSE DRAIN
HOSE CONNECTORS
LINE SAFETIES
BOTTOM GAGE VALVE
MAIN LIQUID
LINE SAFETY
HOSE DRAIN
PRESSURE BUILD
REMOTE CONTROL SHUT OFF
PRESSURE BUILDING COIL
VAPOR RETURN CHECK

Fig.1. Typical piping schematic, pressure unloading trailer

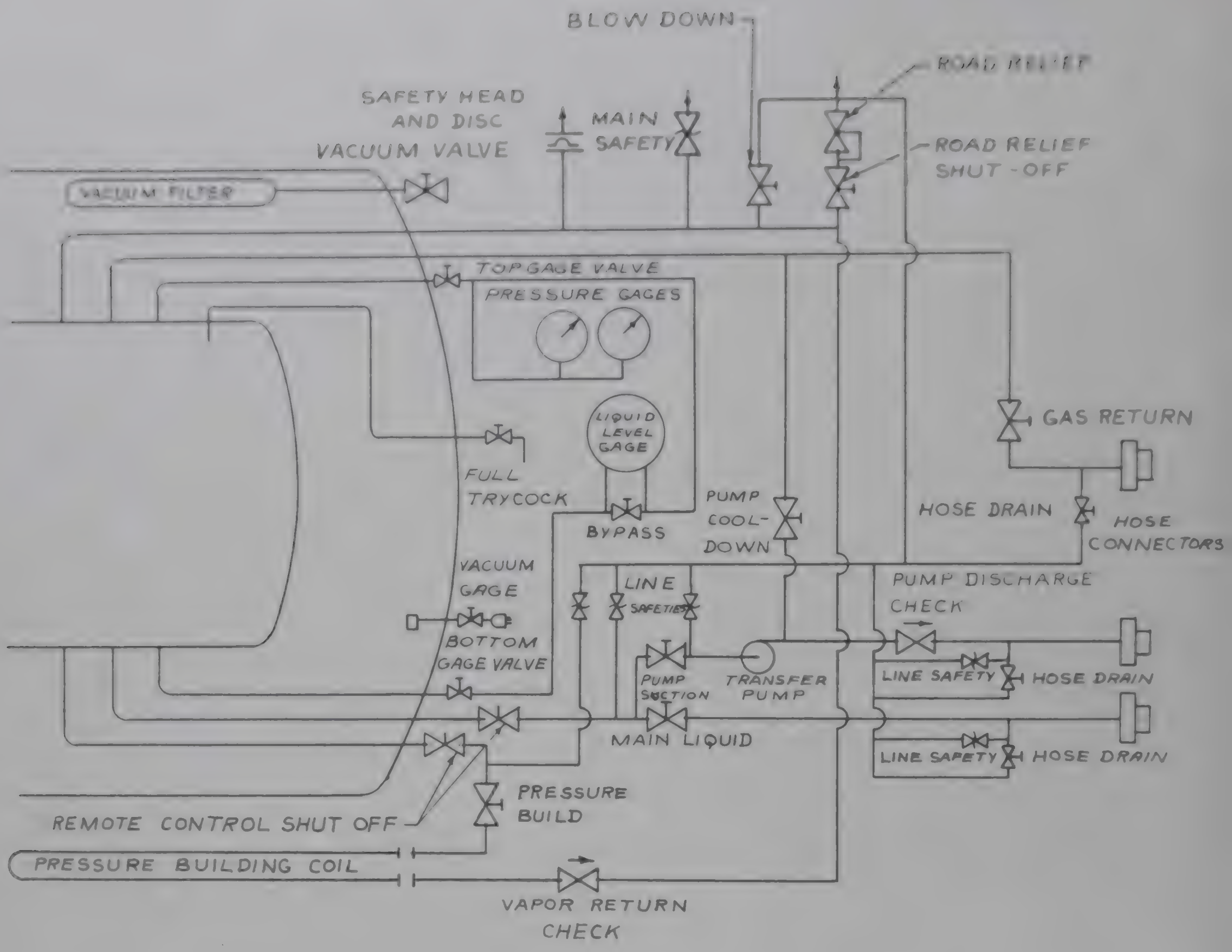
BLOW DOWN
ROAD RELIEF
SAFETY HEAD
AND DISC
MAIN
SAFETY
ROAD RELIEF
SHUT-OFF
VACUUM VALVE
VACUUM FILTER
TOP GAGE VALVE
PRESSURE GAGES
GAS RETURN
LIQUID
LEVEL
GAGE
FULL
TRYCOCK
PUMP
COOL-
DOWN
HOSE DRAIN
HOSE
CONNECTORS
BYPASS
VACUUM
GAGE
LINE
SAFETIES
PUMP DISCHARGE
CHECK
BOTTOM
GAGE VALVE
PUMP
SUCTION
TRANSFER
PUMP
LINE SAFETY
HOSE DRAIN
MAIN LIQUID
LINE SAFETY
HOSE DRAIN
PRESSURE
BUILD
REMOTE CONTROL SHUT OFF
PRESSURE BUILDING COIL
VAPOR RETURN
CHECK

Fig.2. Typical piping schematic, pump unloading trailer

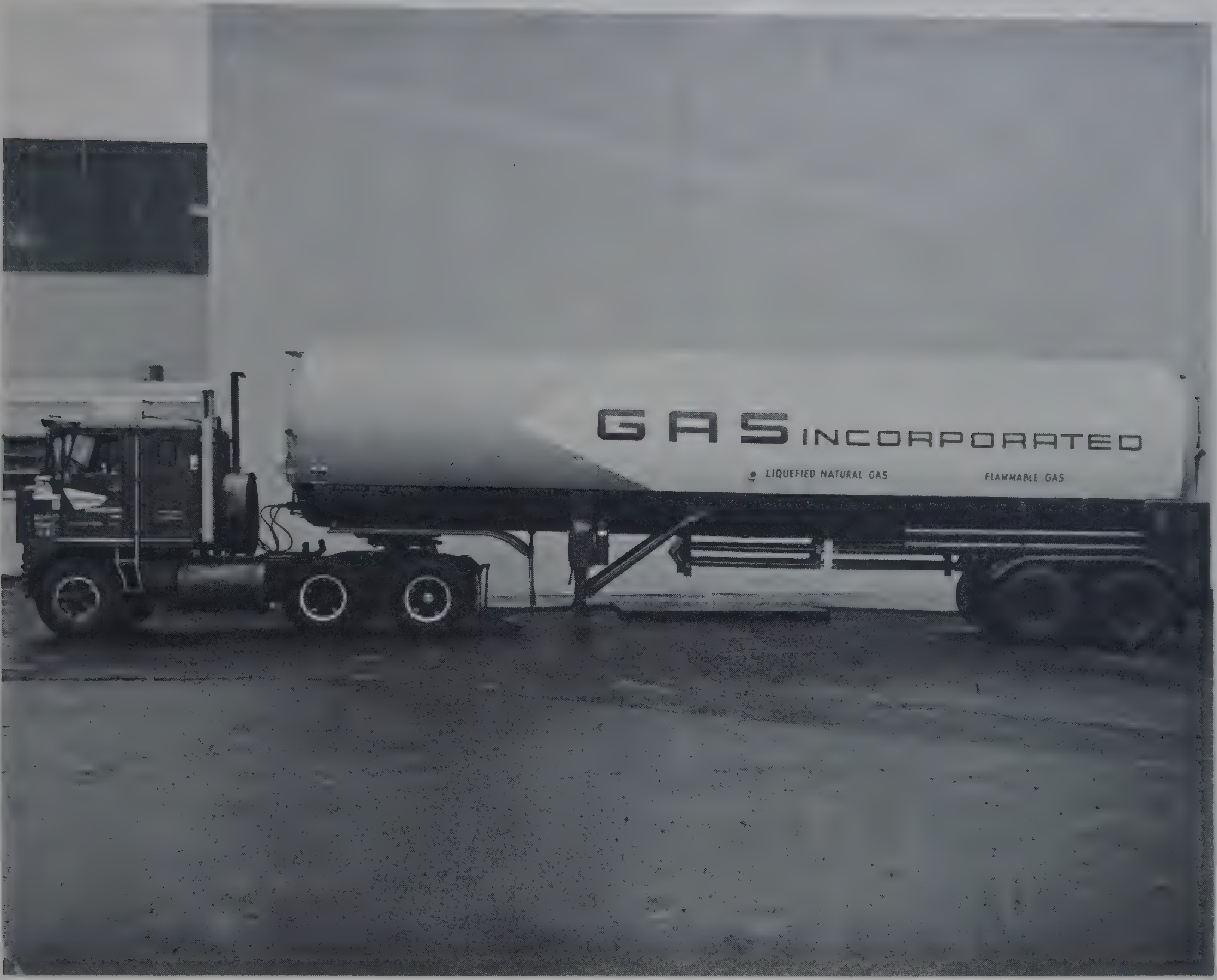
GAS INCORPORATED
LIQUEFIED NATURAL GAS
FLAMMABLE GAS

Fig.3. 11,650 gallon LNG trailer

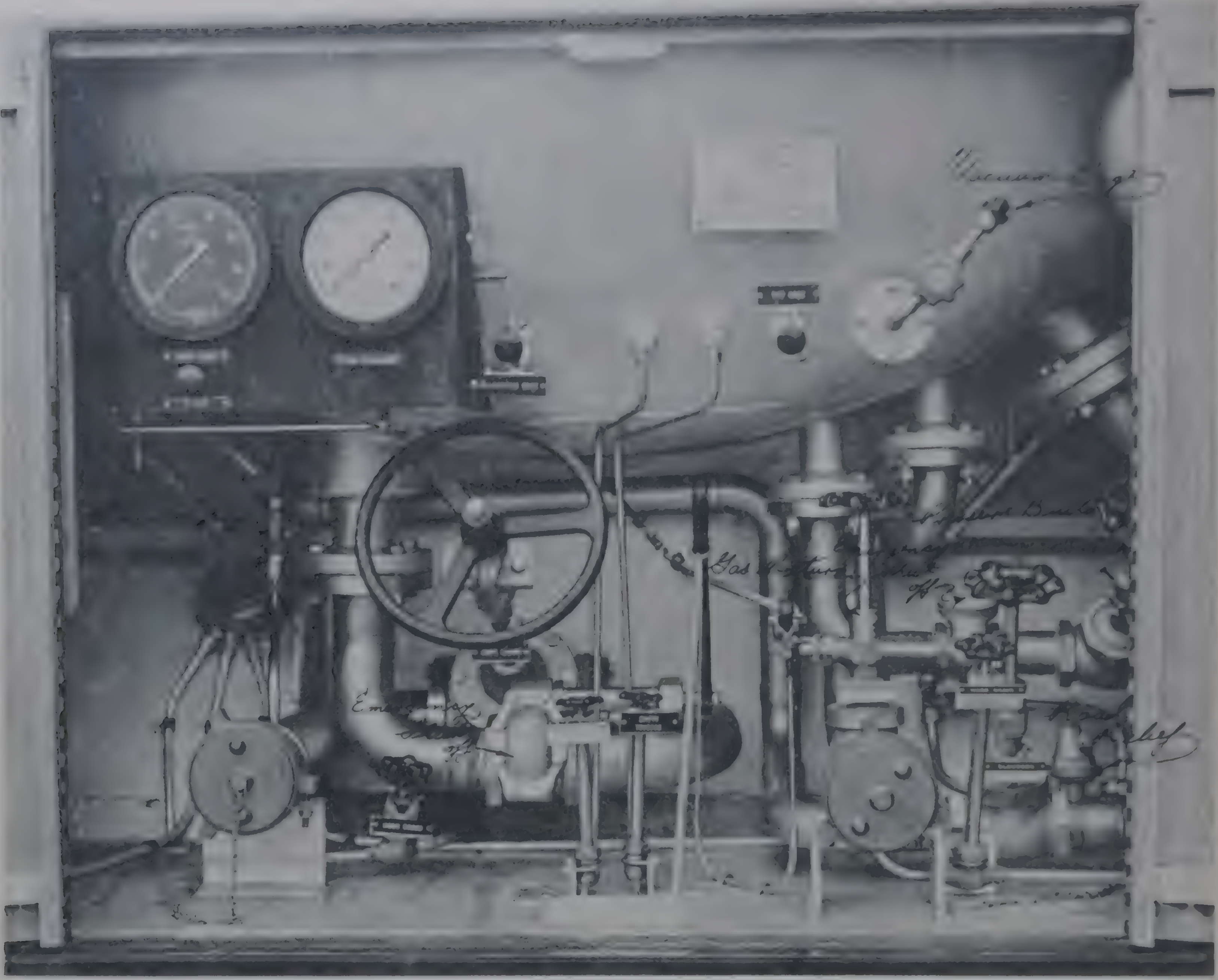
Vacuum Gage
Emergency Shut off
Gas Return Shut off
Emergency Shut off
Road Relief

Fig.4. Control piping on pressure transfer LNG trailer

ECONOMICS OF LNG PRODUCTION AND DISTRIBUTION FOR REMOTE SATELLITE OPERATIONS

By W. D. Bloebaum Jnr.* and J. P. Lewis*

I. INTRODUCTION

In the continental United States today there are fourteen liquefied natural gas plants, either under construction or in operation. All of these, with the exception of the Northern Natural Gas Company plant in Buhl, Minnesota, are large scale peak shaving operations. For these thirteen plants, relatively small amounts of gas are liquefied over a long period of time (200 to 300 days per year) and stored for use during peak winter loads. Typical revaporization rates are 10 to 100 times the liquefaction rate. The primary purpose of liquefaction for peak shaving is storage. This type of operation is generally characterized by returning the gas liquefied to the same pipeline distribution system from which it was originally taken.

Outside the United States, the technology of LNG for base load import is thoroughly developed. In these instances, large quantities of gas are liquefied in an area of low cost gas supply and transported to an area requiring additional amounts of gas. The primary purpose of LNG for base load is to provide lower cost transportation.

Although the techniques involved in the use of LNG for both large scale base load import and peak shaving are well developed, the use of this fuel to supply remote communities with relatively small base load requirements is just beginning to be exploited. This paper reviews in depth the factors which must be evaluated for using LNG for remote applications and the economics involved in these cases. Gas demand considerations, LNG equipment requirements, and economics of an example of such an operation will be covered.

II. SERVICE REQUIREMENTS

The gas service requirements of the community being considered for remote supply with LNG will consist of firm contract sales, interruptible contract sales, and may include electrical power generation. The remote service facilities must be flexible enough to meet the peak demands of all firm contract customers, as well as supply the total usage of this system on a continuous basis. It should be sized such that available interruptible customers will be attracted to the use of natural gas, which in turn will increase total sales and improve the load factor and economics of the remote system.

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*Cosmodyne Corporation

In order to assess properly the equipment needed for remote supply, the peak demand requirements must be determined. Information should be obtained that will generate the design peak hour, design peak day, and design normal month use rates. If the remote system has been previously operating on propane/air or manufactured gas, this data will be readily available. If the system has not yet been piped for gas service, weather bureau records should be consulted. Historical peak demands and total system usage should be reviewed and projected to permit purchase of adequate facilities for future requirements.

It may be well at this point to review the reasons why the use of LNG to supply remote communities should be considered.

1. If the community is already being served with a propane/air or manufactured gas system, the installation of an LNG facility will enable the customers to be converted to natural gas in preparation for pipeline installation. This will provide a developed demand at the time of pipeline completion. Since consumer acceptance of natural gas over propane is well known, a supply from LNG will tend to increase customer preference over all-electric service and provide for better sales potential.

2. If no piped gas service is available, the necessity for preventing the community from going "all-electric" may be urgent. Once appliances are installed for electric service, it becomes extremely difficult to convert these customers to gas in the future.

3. In many cases, the terrain lying between the end of the present transmission line and the remote community is so difficult that pipeline construction is simply not economically feasible. This may be because of a mountain range, a body of water, or distance. If any of these circumstances apply, the transportation of LNG may well prove feasible.

III. EQUIPMENT REQUIREMENTS

The equipment required to service a remote community can be categorized as shown below:

1. Liquefier - A source of liquid natural gas is needed. If the liquid can be purchased from an existing peak shaving facility or base load import terminal, no additional equipment is required. Otherwise, a small liquefier sized to produce the net amount needed for service to the remote community must be installed.

2. Liquefier Storage - This storage should be sufficient to permit continuous operation of the LNG plant while absorbing fluctuations in the community gas demands. This will provide for the most economical operation of the liquefier. In addition, this storage should be adequate to provide continuous gas supply in the event of liquefier maintenance or operating problems. This tank would typically store 5 to 10 days of liquefier output. (See Figure No. 1)

3. Transportation Equipment - The LNG may be transported by semi-trailer, railcar, barge, or combination of these. In most instances, semi-trailer transportation will offer the most flexible means of servicing remote communities. It is quite possible, however, that the use of rail transportation to provide service from a large plant to a supply terminal may be attractive. In the United States, semi-trailers with a capacity of 11,000 gallons (905 MCF) of LNG have been built. (See Figure No. 2) Railcars of approximately 35,000 gallons capacity (2,880 MCF) are also available.

4. Storage at the Remote Site - This storage tank is necessary to provide continuity of supply to the remote community and to eliminate outages during possible interruptions in the overland transportation. The tank should be large enough to absorb peak fluctuations in demand, and to provide some flexibility in the case of bad weather or road closure.

5. Send-out System - This portion of the remote system vaporizes, odorizes, meters and regulates the pressure of the gas delivered to the community distribution system. The equipment is readily available to handle these functions in an unattended manner, with no requirement for external power sources if an ambient air vaporizer is used. This increases the reliability and decreases the cost of operation of such a facility. (See Figure No. 3)

For most cases, back-up equipment would not be required for the major components listed above. As a general rule, this equipment is of an extremely reliable nature. The storage tanks, transportation equipment, and send-out system are such that there are few, if any, moving parts, and supplementary equipment is readily available on a lease basis from cryogenic equipment suppliers. The liquefier can be maintained and repaired during the time provided by storage tank capacity. In the case of an extended shut-down, it would be more economical to transport LNG from an existing plant rather than install redundant capacity solely to support the remote community.

IV. EXAMPLE OF REMOTE COMMUNITY SUPPLY

In order to show the economics involved in supplying LNG to a remote community, a typical example has been chosen. Consider the circumstances where a residential system is now supplied by propane/air. A modest development of the area is planned, thus attracting interest as a potential natural gas market. Since the distance from existing gas distribution lines and difficulty of intervening terrain make justification of a pipeline extension difficult, the use of LNG should be considered. For this example, two sources of LNG are practical: 1) Purchase from a neighboring utility peak shaving plant with a hauling distance of about 140 miles each way, or 2) production of LNG with a small base load plant at the nearest moderate sized community. This would decrease the distance from plant to remote site to 25 miles. The following analysis determines the comparative cost for these two supply methods.

The total fuel requirement under consideration is approximately one-third for the present propane/air customers and two-thirds for electrical power generation. The past use data has been analyzed and indicates the following:

DEMAND ANALYSIS

			MMscf/d
1. Design Peak Day Use			
Peak electrical day	33,000 KWH x 11.5 Mscf/1,000 KWH	.379	
Peak gas day (LPG mixed with air)	$.296 \text{ MMscf} \times \frac{1100 \text{ BTU/scf LPG}}{1070 \text{ BTU/scf Gas}}$	.304	
"Peak-peak" Day			.683
2. Peak Month			
	14.63 MMscf/31 days		.473
3. Average annual usage			
Gas Sales	48.30 MMscf/365 days	.133	
Electrical Power	92.25 MMscf/365 days	.252	
Total	138.98 MMscf/365 days		.385

The above information provides the basis for selecting an appropriately sized liquefaction plant, storage at the liquefier and at the remote site, and selection of proper transportation and send-out equipment. If a small liquefier built specifically for the case being considered is decided, the suggested size is 0.5 MMscf per day. This would provide ample capacity to handle peak month requirements, and enable peak day requirements to be furnished out of liquefier storage. Depending upon the gas supply at the liquefier site, either a conventional non-recycling cascade system or expander process could be considered. For plants of such a small size, it is most likely that a cascade type of liquefaction system would be chosen.

Table I shows an inventory analysis which is provided to assist in selection of appropriate storage tank capacity. Depending on the local road conditions, as well as typical weather profiles, the amount of storage both at the liquefier and at the remote site may be varied. For this example, the suggested storage at the liquefier is a 30,000 gallon (2.5 MMscf) capacity cryogenic tank with a 25 psig maximum working pressure. By holding a reserve semi-trailer filled with LNG at the remote site at all times, additional storage flexibility is provided. This semi-trailer may be used not only for additional storage but also placed in the logistical system for supplying LNG to the remote area. If the LNG is purchased from a neighboring peak shaving plant, storage at that plant would be substituted for the 30,000 gallon tank suggested.

Storage at the remote site has been selected at 60,000 gallons (5MMscf) with a 25 psig maximum working pressure. A normal operating pressure of 10 psig would be sufficient to provide the required 5 psig into the distribution system. As shown by Table I, a combination of storage at the liquefier and at the remote site would permit one week or more of operation under peak load conditions without additional deliveries. Further tank capacity can be chosen if desired by the utility.

The mode of transportation suggested for this example is one 11,000 gallon cryogenic semi-trailer. It is anticipated that the full trailer be hauled to the remote site, the LNG transferred to storage, and the trailer returned empty to the liquefier. The community supply requirements would necessitate a trailer load every three days to meet yearly average requirements and every two days for peak month demand. This is not a rigid transportation schedule considering the one-way distance is only 25 miles. For peak gas requirements, if they occur, more than one trip per day could easily be made.

The cost of LNG service (supply, transportation, storage, and revaporization) to the remote community are categorized as follows:

1. Capital Cost Charges (includes taxes and profits)
2. Operating Cost
3. Commodity Gas Cost

The capital cost schedule is shown on Table II for both producing LNG or purchasing LNG. These costs are based on typical cryogenic equipment prices, which may be obtained from cryogenic equipment manufacturers throughout the world. Although the price shown for the small liquefier may seem modest, it should be remembered that this price does not include storage or typical send-out equipment required for large scale peak shaving systems. In addition, no land costs or site preparation have been included in this figure.

Based on the capital costs just described, Table III presents the cost of service to the remote community during the fifth year of operation. This method of presenting the economics has been chosen in an effort to describe fully the costs involved and to break out the assumptions made so that each utility may modify these costs to fit their individual situation. It is hoped this format will be a guide for evaluation of other remote service projects. Table IV presents similar costs for the case in which LNG is purchased.

In order to decide whether LNG should be produced or purchased, the data from Tables III and IV have been plotted in Figures No. 4 and No. 5. Figure No. 4 shows the cost of gas delivered to the remote system as a function of the cost of purchased LNG. The produced LNG case, of course, is independent of the cost of purchased gas. Figure No. 5 is a more generalized trade-off curve showing a line of equal costs as a function of the LNG purchase price and the difference in distance between a site for production of LNG and the site of an existing LNG plant. This figure readily provides the information as to whether it is preferable to produce or purchase LNG for the example being considered.

It is important to remember that all of the items of equipment described above are easily moved to other locations once pipeline gas is installed to the original site. Since the cost of installation is small, the equipment can be used to "leap frog" ahead of pipeline system development. If the equipment is utilized for more than one remote area, the economics presented will become even more attractive as extended payout periods may be considered.

V. CONCLUSIONS

Based on the data presented above, the following conclusions regarding the use of LNG to supply remote areas can be drawn:

1. The techniques for supplying remote communities with gas via LNG are available and proven. Methods of liquefaction, storage, transportation and send-out have all been demonstrated previously. The use for remote supply is merely a logical extension of developed technology.

2. The economics involved in gas service to remote communities with LNG are attractive. As a comparison, the following are break-even costs (on an equal BTU content) for LPG and diesel fuel:

BREAK-EVEN COSTS

LPG - \$. 0374 per delivered, processed gallon

DIESEL - \$. 1118 per delivered gallon

The prices shown are not contract purchase prices, but also must include all appropriate charges for delivery, storage, mixing and capital return analogous to those found in Tables III and IV.

3. By providing gas supply to a newly developing community, take-over by electrical service is prevented. This in itself may well justify absorbing higher than normal costs of service, since it will "hold" the community for future gas business.

4. Even if LNG is not presently available for purchase from an existing peak shaving plant, it is economical to build a small liquefaction facility for the sole purpose of serving a remote community. These economics were shown in Tables III and IV.

It is hoped that the above information will be useful to any utility company contemplating serving a remote community with LNG. The format of the economics presented should lend themselves to modification for individual situations. Although this concept of remote community supply is relatively untried, the successful history of LNG in base load import and large scale peak shaving has provided the necessary experience and learning for successful operations of this type.

TABLE I

LNG INVENTORY ANALYSIS

	Capacity MMscf	Probable Peak Season Inventory MMscf	Peak Day Equivalent Days*	Peak Month Equivalent Days*	Yearly Average Equivalent Days*
Remote Site					
60,000 Gallons	5.0	4.0	5.9	8.5	10.5
Liquefier					
30,000 Gallons	2.5	2.0	2.9	4.2	5.2
TOTAL	7.5	6.0	8.8	12.7	15.7
With the ability to hold a tanker at the remote site the inventory may be increased as follows:					
Remote Site					
60,000 + 11,000 gallons	6.0	5.0	7.3	10.6	13.1
Liquefier					
30,000 Gallons	2.5	2.0	2.9	4.2	5.2
TOTAL	8.5	7.0	10.2	14.6	18.3

* Based on Probable Peak Season Inventory.

TABLE II

CAPITAL COST SCHEDULE

Produced LNG

Liquefier	$ 329,000
Liquefier Storage	47,000
Remote Site Storage and Vaporization	78,000
Tanker	42,000
TOTAL:	$ 496,000

Purchased LNG

Remote Site Storage and Vaporization	$ 78,000
Tanker	42,000
TOTAL:	$ 120,000

TABLE III

COST OF SERVICE, 5TH YEAR

PRODUCED LNG

Capital Cost Charges (5th Year)			
Depreciation: (20 yrs. straight line)	.0500		
Profit: $.065 \times [1.00 - (4 \times .05)]$ =	.0520		
Property Taxes: $.025 \times (.80)$ =	.0200		
Income Taxes: $.065\,(.80)\,\frac{.52}{.48}$ =	.0562		
Insurance: $.025 \times (.80)$ =	.0200		
	.1982		
$.1982 \times \$496,000$			$ 98,307
Operating Costs			
Liquefier Power Costs			
$325\text{ hp} \times \frac{\$.005}{\text{hp-hr}} \times 8760\,\frac{\text{hr.}}{\text{yr.}} \times \frac{.383}{500}$ load factor =		10,904	
Liquefier Operation and Maintenance		10,000	
Tanker Operation			
$15/trip x 153 trips		2,295	
$25\,\frac{\text{mi.}}{\text{trip}} \times 2\text{ ways} \times \$.43/\text{mi.} \times 153\text{ trips}$		3,289	26,488
Total Cost above commodity cost			$124,795
Cost Mscf $124,795/139,000 Mscf/yr			.8978
Estimated Gas Commodity Cost: $.33/Mscf			.3300
Total Delivered Gas Cost, 5th Year, $/Mscf			1.2278

Note: This includes $25,792 profit which is equivalent to $.1855/Mscf of the above cost.

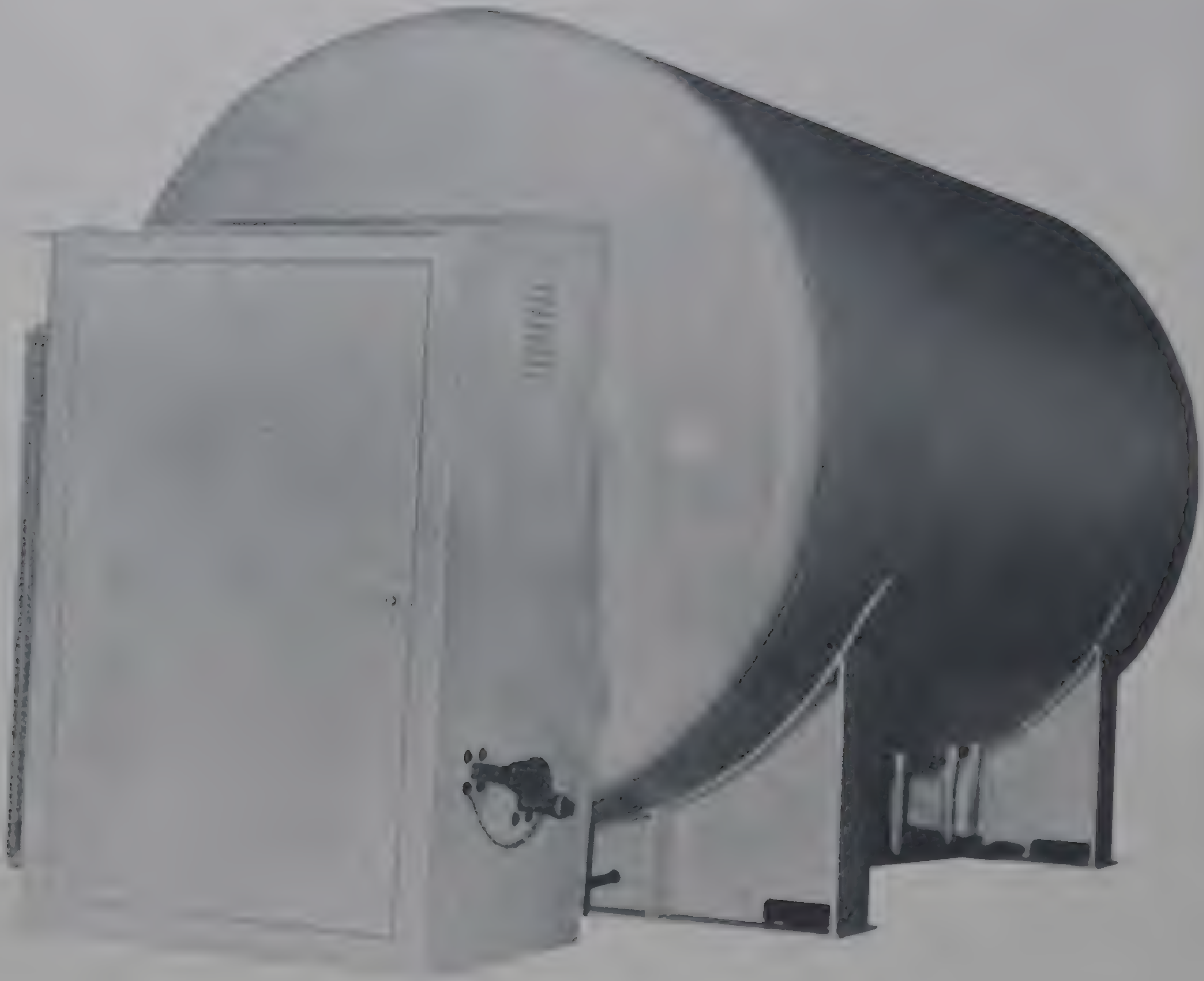
TABLE IV

COST OF SERVICE, 5TH YEAR

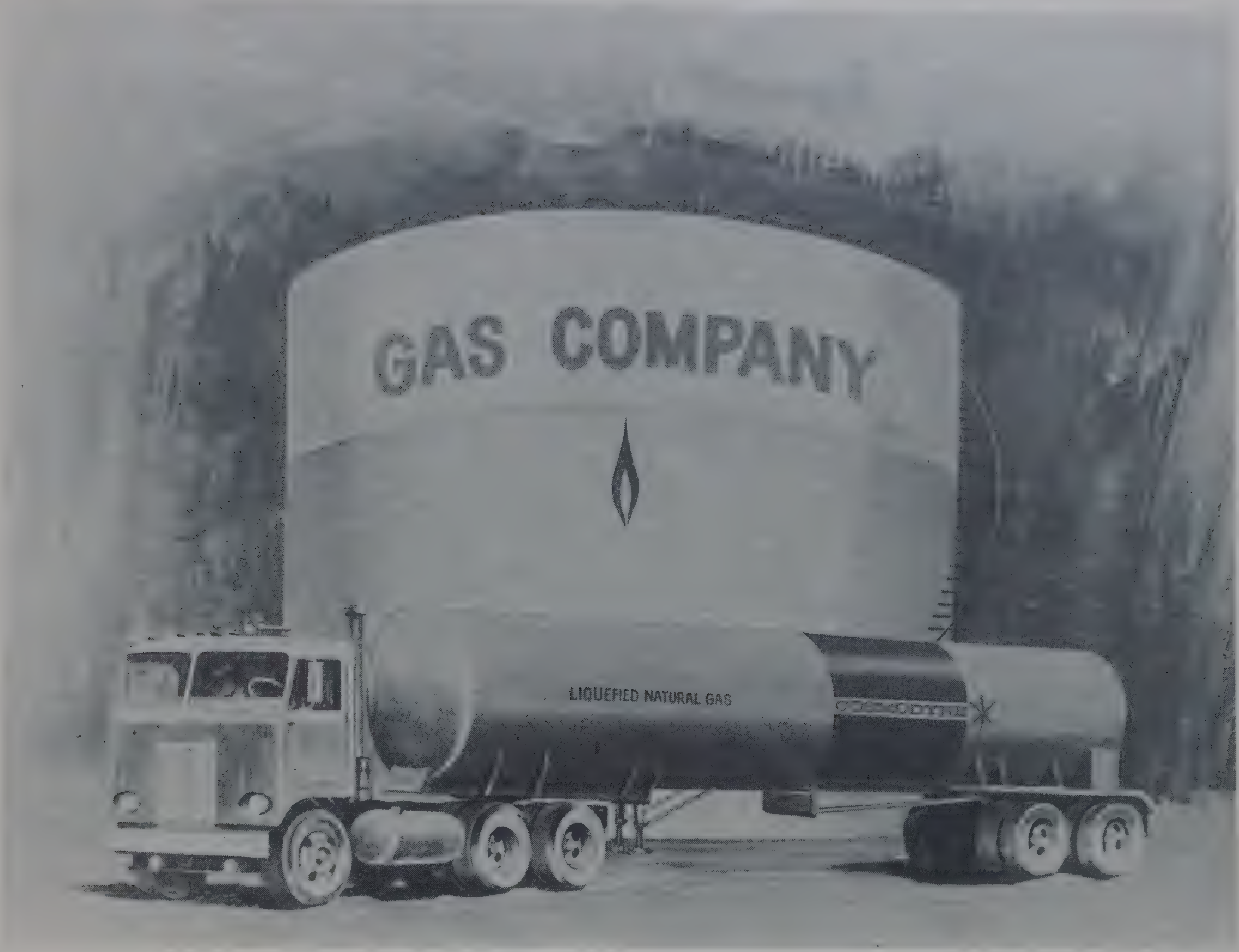
PURCHASED LNG

Capital Cost Charges		
Depreciation: (20 yrs, straight line)	.0500	
Profit: .065 x [1.00 - (4 x .05)] =	.0520	
Taxes: .025 x (.80) =	.0200	
Income Taxes: .065 (.80) $\frac{.52}{.48}$ =	.0562	
Insurance: .025 x (.80) =	.0200	
.1982 x $120,000	.1982	$23,784
Operating Costs		
Tanker Operation		
$15/trip x 153 trips	2,295	
140 $\frac{\text{miles}}{\text{trips}}$ x 2 ways x $.43/mi x 153 trips	18,421	
		20,716
Total Cost above commodity cost		$44,500
Cost/Mscf $44,500/139,000 Mscf/yr		.3201
Estimated cost of LNG $\frac{8\ 1/2\ \text{cents/gal.}}{82.3\ \text{ft}^3\ \text{gal.}}$ x 100 ft^3/Mscf		1.0328
Total Delivered Gas Cost, 5th Year, $/Mscf		1.3529

Note: This includes $6,240 profit which is equivalent to $.0449/Mscf of the above cost.

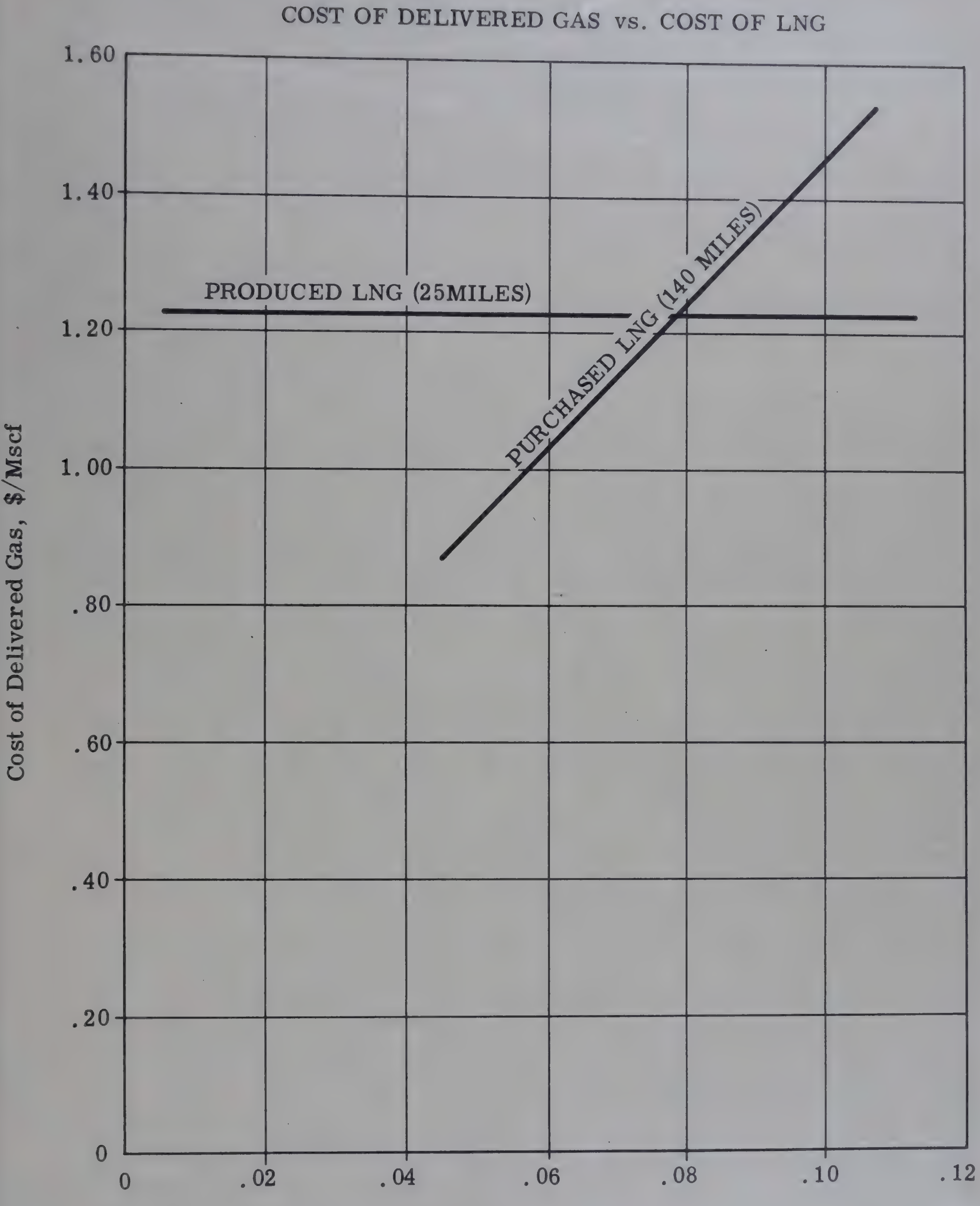


1. LNG Storage Tank

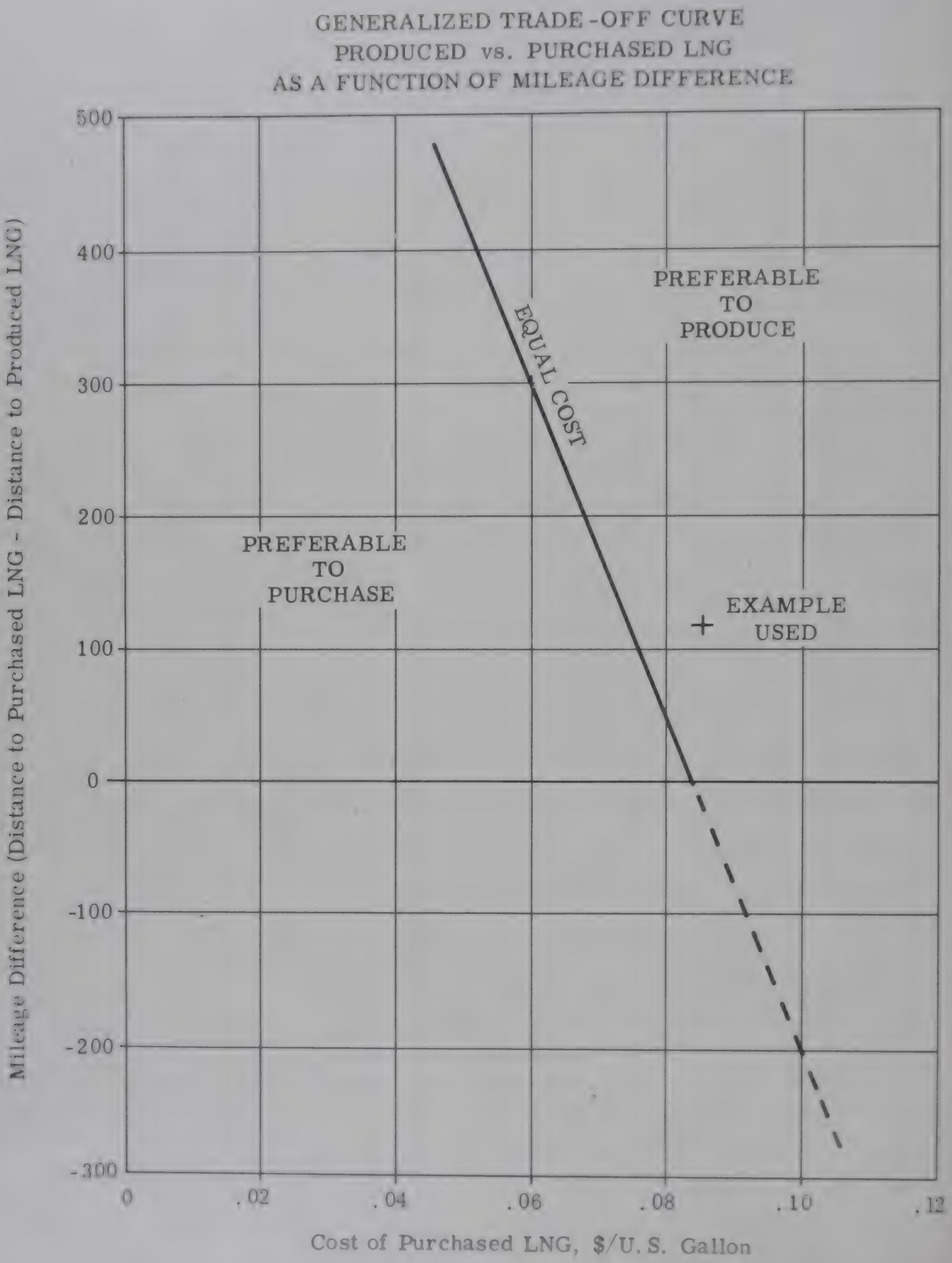

GAS COMPANY
LIQUEFIED NATURAL GAS
COSMODYNE

2. LNG Semi-trailer, 11,000 gallon capacity

3. Typical remote LNG facility showing storage tank and send-out system



4. Cost of Delivered Gas vs. Cost of LNG



5. Generalized Trade-off Curve

TERMINAUX DE G.N.L.
DESCRIPTION DE PROCEDES DE VAPORISATION ET DES MATERIELS CORRESPONDANTS

By J. P. Buffière

A la suite des nombreuses études effectuées par la société "Air Liquide" sur le problème de la revaporisation de gaz naturel liquéfié dans un terminal, l'auteur fait le point :

- sur les procédés nouveaux proposés pour le traitement du GNL
- sur les matériels de vaporisation nécessités par la mise en oeuvre de ces procédés.

I - CONSIDERATIONS GENERALES SUR LE TRAITEMENT DU G.N.L.

Le traitement du GNL dans un terminal dépend de plusieurs facteurs :

1°) Composition du GNL à la revaporisation (Annexe N° 1)

Ci-joint un tableau des compositions moyennes des principaux projets de GNL réalisés ou en cours d'étude. Il s'agit de compositions au terminal.

On voit la gamme très étendue des GNL étudiés.

2°) Spécifications du gaz revaporisé

Le gaz naturel revaporisé doit répondre à différentes spécifications suivant l'utilisateur qui peut être :

a) -soit un gros particulier (usine chimique, usine électrique etc..

b) -soit un gazoduc servant de réseau de distribution dans le pays considéré.

Dans ce cas le gaz revaporisé doit pouvoir être compatible avec le gaz commercialisé déjà existant.

Nous donnons (annexesN° 2 et N° 3) à titre d'exemple, le diagramme d'interchangeabilité montrant qu'en France le gaz naturel pour être commercialisé doit se trouver dans une certaine zone du diagramme (indice de Wobbe - potentiel de combustion).

D'autre part le gaz revaporisé doit s'adapter au pouvoir calorifique supérieur (PCS) du gaz déjà commercialisé, par exemple en France 9,7 th/Nm3. ± 1 %.

3°) Pression de livraison du gaz

Elle varie évidemment suivant l'emplacement du terminal par rapport aux utilisateurs.

Dans le cas d'un réseau de distribution, la pression à assurer se situe en général entre 50 et 100 bars, donc dans la zone hypércritique du gaz.

4°) Débouchés possibles pour les constituants plus lourds que le méthane

On constate que les GNL contiennent des teneurs en C2, C3, C4 et C5 + non négligeables. Il peut être intéressant d'extraire ces produits si l'on peut les valoriser dans des usines pétrochimiques. Les LPG (C3, C4) sont en général commercialisés, quant à l'éthane il est une matière première noble pour l'industrie de l'éthylène et de ses dérivés.

ANNEXE N°1

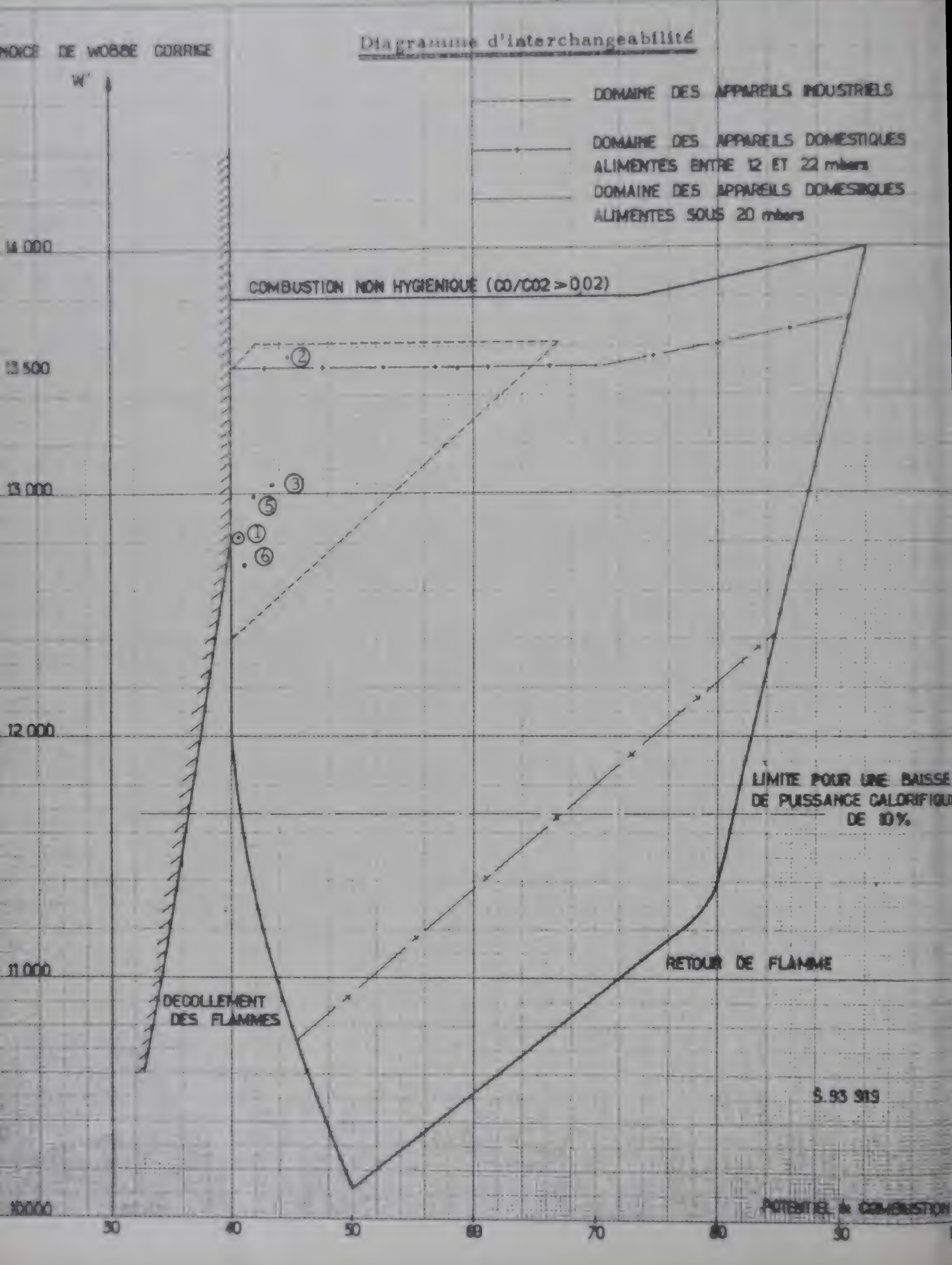
COMPOSITION MOYENNE DE DIFFERENTS G.N.L.

(à la revaporisation)

Numéro		1	2	3		4	5	6
Provenance		Lacq	Arzew	Skidda		Libye	Alaska	Sahkaline
Terminal		Nantes	Le Havre Canvey	Fos/Mer		Barcelone	Japon	Japon
Composition % volume	N2	0,2	0,7	0,9	1,0	-	0,6	1,0
	C1	98,8	87,9	87,2	92,8	67,7	95,6	97,4
	C2	0,9	8,1	8,2	5,4	19,8	2,3	1,5
	C3	0,1	2,4	2,5	0,6	8,7	1,3	0,1
	iC4	-	0,4	0,4	0,1	1,5	0,1	-
	nC4	-	0,4	0,6	0,1	1,6	-	-
	C5 +	-	0,1	0,2	-	0,7	-	-
PCS th/Nm3		9,60	10,62	10,69	9,97	13,15	9,85	9,56
W'		12 830	13 540	13600	13040	15 340	12 980	12 720
C		40,8	45,1	45,2	43,2	65	41,9	40,7

ANNEXE N°2 ④

Diagramme d'interchangeabilité



ANNEXE N°3

Diagramme d'interchangeabilité des gaz de 2ème. famille

Indice de Wobbe corrigé W'

$$W' = K_1 , K_2 \frac{PCS}{d}$$

d = densité par rapport à l'air

PCS = pouvoir calorifique supérieur en Kcal/Nm3

K_1 = coefficient de correction tenant compte de la présence d'hydrocarbures dans le gaz autre que le méthane

K_2 = coefficient correctif égal à 1 quand les teneurs en CO, O_2 et CO_2 sont nulles.

Potentiel de combustion C

$$C = u \times \frac{H_2 + 0,3\ CH_4 + 0,7\ CO + v\ a\ C_nH_m}{d}$$

u et v coefficients

H_2, CH_4, CO, C_nH_m = teneurs en % volume

a = potentiel de combustion (a = 0,95 pour C_2H_6 et C_3H_8)

II - PROCEDES NOUVEAUX DE TRAITEMENT DE G.N.L.

Généralités sur la récupération des frigories

Les frigories disponibles dans le G.N.L. sont de l'énergie potentielle. La question est de savoir à quel prix elle peut être récupérée. Rappelons qu'il a fallu dépenser environ 0,36 kWh par Nm3 de gaz naturel pour le liquéfier.

Jusqu'à présent la récupération économique maximum a été celle obtenue en donnant au G.N.L. sous forme liquide la pression élevée (50 à 100 bars) nécessaire à son transfert par gazoduc. Cette récupération est de l'ordre de 30 % par rapport à une compression sous forme gazeuse.

compression à 70 bars : 0,12 kw

énergie nécessaire pour regazéification - 0,01

soit $\frac{0,11}{0,36} \simeq 30\ \%$ d'énergie récupéré

Une autre idée est la récupération d'énergie par un cycle intermédiaire à propane et turbine de détente récupérant l'énergie.

environ 0,014 kW/Nm3

c'est-à-dire environ 5 % d'énergie récupérée.

Il est décrit ici des procédés nouveaux permettant la valorisation des frigories disponibles dans le G.N.L. afin de résoudre économiquement le problème de la livraison de gaz commercial répondant à des spécifications données.

1- Production de gaz commercial à PCS plus faible que le GNL en effectuant un dégazolinage sur le gaz naturel

Suivant le PCS initial du GNL et l'intérêt éventuel des produits lourd extraits, on peut avoir recours à une extraction complète ou non.

a) Extraction totale

Dans le cas où l'éthane est valorisable il est intéressant d'employer le schéma suivant

Le gaz d'émission est produit en deux étapes

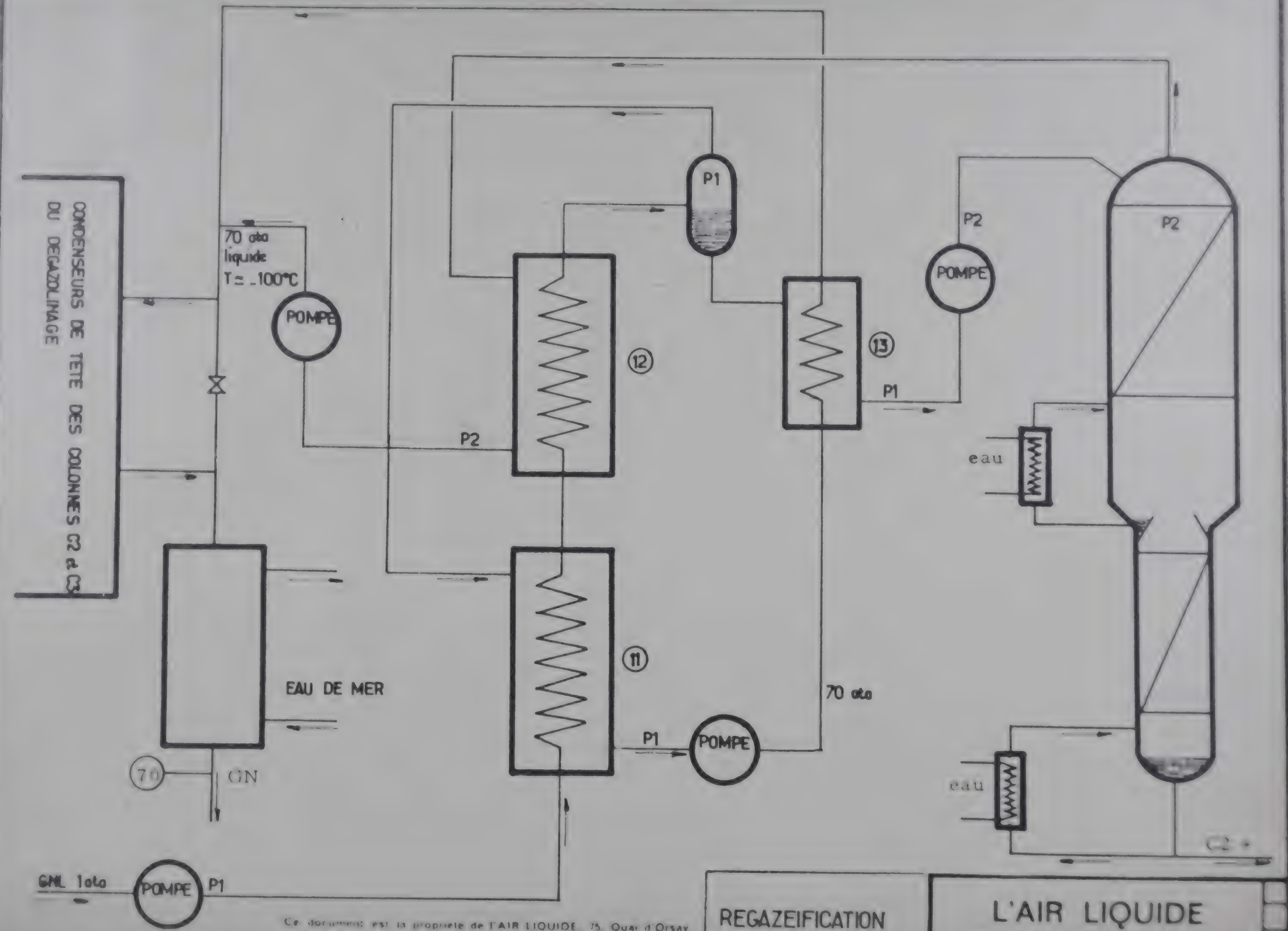
- l'un produit dans le séparateur, est reliquéfié dans l'échangeur 11
- l'autre produit en tête de la colonne de fractionnement C1/C2 + est reliquéfié à plus haute pression dans l'échangeur 12.

La totalité du gaz d'émission est donc reliquéfiée après l'enlèvement de la fraction C2 +. Cette reliquéfaction permet donc sa compression à 75 bars (pression du réseau) sous forme liquide donc à dépense d'énergie minimale.

La valorisation des frigories froides (-160° à - 75°C) est ainsi totalement effectuée sur l'unité de traitement du GNL qui assure :

- la reliquéfaction totale du gaz d'émission produit
- les besoins frigorifiques (entre - 110 et - 75°) des colonnes de dééthanisation et dépropanisation de l'installation de dégazolinage
- la livraison d'éthane, propane et butane liquides sous-refroidis jusqu'à leur température d'ébullition à pression atmosphérique pour être stockés.
- la livraison éventuelle des frigories intéressantes restantes sur le GNL (-75° -40°) par un circuit indépendant de propane pouvant les restituer par exemple à une unité d'éthylène associée.

Le procédé assure ainsi la possibilité d'obtenir des rendements d'extraction en éthane ($\geqslant$90 %) et LPG ($\geqslant$95 %) très élevés avec une consommation d'énergie minimale (de l'ordre de 0,01 kW par Nm3/h regazéifié) analogue à une simple regazéification en utilisant toujours des pompes de liquides.


CONDENSEURS DE TETE DES COLONNES C2 et C3
DU DEGAZOLINAGE
70 ata
liquide
T = -100°C
POMPE
P2
⑫
P1
⑬
P1
P2
POMPE
P2
eau
eau
C2+
⑪
70 ata
P1
POMPE
EAU DE MER
70
GN
GNL 1ata
POMPE
P1

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REGAZEIFICATION de GNL

L'AIR LIQUIDE

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Une autre caractéristique du procédé réside dans la possibilité d'utiliser des pressions pour les colonnes de fractionnement permettant d'employer systématiquement des chauffages par de l'eau, ce qui réduit les frais annuels d'exploitation au minimum(uniquement énergie de pompage de l'eau de mer).

b) Extraction moins complète de C2 +

Dans le cas où l'on ne veut extraire qu'une partie de l'éthane (rendement $\leqslant 60$ %), un schéma plus simple et à moindres investissements peut être utilisé.

c) Extraction des LPG seuls

Si l'on n'a pas la possibilité de valoriser l'éthane, on peut n'extraire que les LPG.

Dans ce cas, il suffit de traiter le GNL à une pression P 1 et de recondenser à cette même pression le gaz d'émission dégazoliné.

On a ensuite une colonne de dégazage du méthane et éthane restant dans les produits lourds (C3 +). Ce "fuel" peut être recondensé et renvoyé à l'aspiration d'une pompe pour le renvoyer dans le réseau.

d) Extraction des LPG avec injection d'azote

La récupération même maximum des hydrocarbures plus lourds que l'éthane (C3 +) contenus dans le gaz naturel peut ne pas être suffisante pour assurer le PCS du gaz émis.

Afin d'obtenir un PCS notamment plus faible (de l'ordre de 9,7 th/Nm3 par exemple), il est alors nécessaire de réinjecter dans le gaz émis une fraction gazeuse volatile tel que azote ou air. Le procédé couramment utilisé était de comprimer à l'état gazeux ce fluide à la pression élevée du réseau (par exemple 70 bars) et de le mélanger au gaz émis, ce qui nécessite un compresseur ayant un gros taux de compression, donc de gros investissements, une grosse dépense d'énergie et des difficultés sérieuses d'exploitation.

Un procédé a été mis au point qui permet de limiter considérablement l'énergie et les investissements correspondant à cette injection d'azote.

A titre indicatif, en considérant le GNL d'Algérie ayant un PCS de 10,7 th/Nm3, pour rendre le gaz substituable avec le gaz de Lacq (PCS = 9,7 th/Nm3), il est indispensable après avoir fait l'extraction des C3, C4 et C5 de réinjecter environ 4 % d'azote dans le gaz. Pour une installation traitant environ 450 000 Nm3/h, la réduction de consommation d'énergie par rapport à la solution classique est de 50 à 60 %.

2 - Production de gaz commercial sans extraction de produits lourds par injection d'azote

Dans le cas où le traitement du GNL n'implique pas la séparation de produits lourds, un procédé a été mis au point qui permet de réduire considérablement la dépense d'énergie et les investissements correspondant à cette injection.

A titre d'exemple, pour un GNL algérien de PCS 10,3 th/Nm3, il faut injecter environ 6 % d'azote dans le GNL pour qu'il ait le PCS de 9,7 th/Nm3 ; la réduction de dépense d'énergie par rapport à la solution classique est d'environ 70 %.

3- Vaporisation simple de LNG avec reliquéfaction simple des évaporations des stockages

Si les caractéristiques du GNL sont telles qu'une simple vaporisation est suffisante, on peut de la même façon utiliser la chaleur sensible disponible sur le GNL pour reliquéfier à basse pression les gaz d'évaporation des réservoirs de GNL. Ceux-ci sont injectés dans le GNL et ramenés à l'état liquide à la pression d'aspiration des pompes de compression portant le gaz naturel à la pression finale du réseau.

Exemple :

débit GNL vaporisé 450 000 Nm3/h

2 réservoirs de 50 000 m3 liquide

débit évaporation 0,1%/jour soit 2 500 Nm3/h

Les évaporations peuvent être reliquéfiées avec une réduction de dépense d'énergie de 85 % par rapport à la solution classique.

4- Production simultanée d'azote et d'oxygène liquide

L'intérêt de l'utilisation des frigories contenues dans le GNL pour la production d'oxygène ou d'azote est variable en fonction des opérations envisagées :

- l'intérêt est faible quand il s'agit seulement de produire de l'oxygène et de l'azote gazeux car la tenue en froid des installations "Oxytonne" représente seulement 10 % de la dépense d'énergie globale de l'installation.

- Par contre l'utilisation des frigories du GNL pour la production d'oxygène et d'azote liquide peut être très intéressante dans la mesure où cette utilisation se limite à la partie "noble" des frigories du GNL, c'est-à-dire à la frange -140° -155°C.

La dépense d'énergie spécifique est abaissée d'environ :

45% dans le cas de l'oxygène liquide

60% dans le cas de l'azote liquide.

D'autre part, dans un cas analogue à celui du paragraphe 1 ci-dessus (production de G.C. avec extraction d'éthane et de LPG), il est possible de produire sans difficultés 5 000 l/h d'oxygène et 5 000 l/h d'azote liquide quand on traite 1,5 10^9 Nm3/an de gaz naturel. La seule conséquence est une diminution du rendement d'extraction en éthane qui passe de 90 à 80 % environ, les rendements en LPG restant inchangés.

Les différents procédés décrits ci-dessus ont fait l'objet de brevets de la société l'Air Liquide.

II - DESCRIPTION DE MATERIELS DE VAPORISATION

Dans tous les procédés décrits plus haut nous avons vu la nécessité de vaporiser des liquides en grande quantité jusqu'à une température voisine de la température ambiante. En particulier on pourra remarquer que les pressions des colonnes sont choisies de façon à avoir toujours en cuve des températures voisines de 0°C. Le lieu du terminal se trouvant par définition près de la mer, on dispose d'une source de chaleur très économique et en quantité illimitée, les frais d'exploitation se réduisent alors à l'énergie de pompage de l'eau de mer.

Il est décrit ici, des matériels de vaporisation à hautes performances thermiques basés sur le principe du ruissellement d'eau de mer apportant ses calories sur la paroi extérieure de tubes verticaux.

1°) Echange côté ruissellement d'eau

Soit un tube vertical pourvu ou non d'ailettes extérieures longitudinales. L'eau est distribuée à la partie supérieure à l'aide d'un distributeur qui répartit l'eau de façon régulière autour du tube ou des ailettes, et s'écoule par gravité sur toute la hauteur du tube.

Nous avons expérimentalement constaté que l'on pouvait obtenir un écoulement régulier de l'eau (c'est-à-dire que la surface d'échange extérieure est complètement mouillée) pour des débits d'eau très importants. Avec une disposition particulière du distributeur, il n'y a pas décollement du film d'eau sur le tube comme on aurait pu le penser pour des débits d'eau importants.

On obtient ainsi des coefficients d'échange (K_2) eau-paroi supérieurs à ceux obtenus par circulation forcée d'eau dans un tube. On a pu mesurer également le coefficient d'échange eau-glace évidemment bien inférieur, mais encore très acceptable.

2°) Echange côté fluide à vaporiser

Le fluide froid en ébullition (ou réchauffement) échange ses frigories avec la paroi en se vaporisant de bas en haut à l'intérieur du tube refroidi par l'eau.

Pour réaliser cet échange, il faut tenir compte des deux considérations suivantes :

- ne pas apporter à l'intérieur du tube plus de frigories que la paroi n'est capable d'en évacuer côté eau

- apporter ces frigories le plus uniformément possible en tous points de la paroi intérieure du tube afin d'utiliser au mieux la surface d'échange extérieure en maitrisant la formation de glace.

Afin d'augmenter l'échange côté fluide, il faut aménager l'intérieur du tube pour arriver, d'une part à freiner l'échange dans la zone très froide (pour maîtriser la formation de glace qui limiterait le débit) et d'autre part à l'accélérer au maximum dans la zone ambiante (pour réduire la hauteur du tube).

Pour arriver à ce résultat nous avons employé différentes surfaces de travail :

a) - tube extérieur
    - tube lisse
    - tube à ailettes extérieures longitudinales

b) remplissage intérieur

Différents aménagements intérieurs du tube ont été mis au point technologiquement et expérimentés dans les conditions de service.

Le choix de ces surfaces de travail dépend de la nature du fluide à vaporiser, de la pression, température et perte de charge admissible et évidemment de critères économiques.

3°) Calcul sommaire du tube vaporiseur

On peut rapidement procéder au dimensionnement de la façon suivante :

Pour un diamètre intérieur et un profil du tube extérieur donnés, le débit maximum de fluide à vaporiser est donné par l'équation suivante au bout froid.

$$K_1 S_1 (To - T_1) = K_2 S_2 (T_2 - T_o)$$

K_2, S_2, T_2, T_1, S_1 et T_o (= 0° pour éviter la glace) sont connus

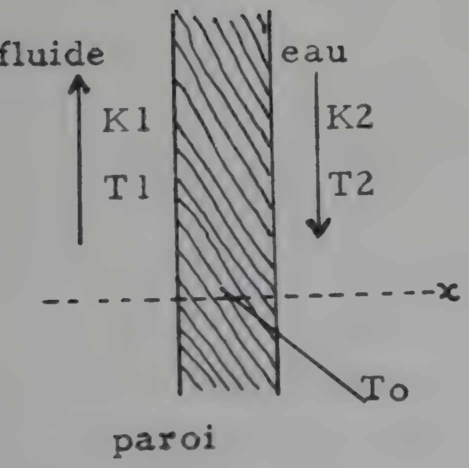
on en déduit $K_1 = A \times \frac{G_1^{\,0,8}}{d_1^{\,0,2}}$ donc G_1

A = coefficient physique du fluide

d_1 = diamètre hydraulique $= \frac{4 \times \text{section}}{\text{périmètre mouillé}}$

G_1 = débit massique en kg/h/cm2 $= \frac{\text{débit poids}}{\text{section de passage}}$

G_1 est donc déterminé



On conduit ensuite le calcul de la façon suivante :

A l'abscisse x , sur un tronçon de tube de longueur dx , on échange une quantité de chaleur

$$dQ = K_1 dS_1 \ \Delta T_1 = K_2 dS_2 \ \Delta T_2$$

On ne peut pas jouer sur le coefficient K_2 (eau-métal) qui est pratiquement constant. Pour avoir le dQ maximum, on augmente T_2 en abaissant la température de paroi aux alentours de 0°C.

Connaissant G_1, on a alors K_1, donc la section de passage nécessai à l'abscisse x.

On arrive ainsi à obtenir des coefficients d'échange très élevés conduisant à des hauteurs de tube pas trop élevées (de l'ordre de 4 à 6n

Il faut naturellement vérifier que la perte de charge n'est pas trop élevée, ce qui peut conduire à une limitation du débit par tube.

4°) Description sommaire du vaporiseur

Le vaporiseur est constitué par l'ensemble de ces tubes verticaux assemblés par exemple en rideaux et supportés de façon à constituer un ensemble compact facilement transportable et montable.

Il comporte un collecteur d'eau, un collecteur de liquide à vaporiser et un collecteur de gaz.

Une cuvette recueille l'eau de ruissellement ayant servi à la vaporisation.

5°) Sécurité de fonctionnement

Ce type de vaporiseur a été choisi de préférence à une solution classiq (calandre et tubes) afin d'éviter les risques d'éclatement des échangeurs par gel au moment des démarrages et arrêts de l'installation ou par suite de fausses manoeuvres sur le circuit d'eau.

6°) Résultats d'essais et applications pratiques

Les calculs et hypothèses faites ont été vérifiés par des essais systématiques de ces matériels essayés dans les deux cas suivants :

a) Vaporisation (réchauffage) à haute pression (entre 20 et 100 bars) de fluides à bas point d'ébullition N2, CH4, C2H4.

Nous avons obtenu des courbes donnant le débit maximum pouvant être vaporisé par un tube donné en fonction de la température d'eau employée, de l'écart au bout chaud et de la perte de charge admise. Ce type de vaporiseur convient très bien pour la vaporisation de GNL à haute pression.

b) Vaporisation à plus basse pression (entre 1 et 5 bars) de fluides ayant un point d'ébullition voisin de la température de l'eau

Nous avons essayé ce type de vaporiseur comme rebouilleur de cuve de colonne de dégazolinage de gaz naturel.

Le liquide à vaporiser était envoyé à la partie inférieure du tube à partir de la cuve de la colonne de distillation. Il s'écoulait par gravité, le mélange liquide-vapeur en sortie du tube vaporiseur retournant à la cuve de la même colonne.

Dans certains cas d'application le liquide à vaporiser a son point d'ébullition dans les conditions de pression de la colonne très voisin de la température de l'eau mise à notre disposition ; Il importe alors que la pression en cours de vaporisation soit maintenue aussi basse que possible afin de ne pas relever de façon excessive la température de vaporisation.

Il était donc nécessaire que la circulation du mélange liquide-vapeur en cours de vaporisation se fasse avec une perte de charge faible (pas de pompe de reprise du liquide, charge hydraulique faible).

Les essais ont vérifié essentiellement les deux points suivants :

- l'échange thermique entre le fluide vaporisé et l'eau
- l'écoulement vertical de bas en haut du fluide en vaporisation par simple gravité.

L'élément de vaporisation était constitué par un tube en aluminium lisse à l'extérieur et ayant des ailettes intérieures obtenues par extrusion

Le fluide vaporisé était du butane commercial à la pression atmosphérique.

On a vérifié dans ces conditions que l'on pouvait faire varier le taux de recyclage du liquide de 0 à 60 % en agissant sur le débit d'eau de ruissellement sans qu'il y ait par ailleurs distillation dans le tube (accumulation de produits lourds)

CONCLUSION

Les terminaux de réception de gaz naturel liquéfié peuvent présenter des problèmes très divers en fonction

- du gaz naturel liquéfié traité
- de l'intérêt économique de certaines extractions
- de la pression et des caractéristiques recherchées pour le gaz commercia
- de l'intérêt économique de certaines récupération de frigories.

Des études thermodynamiques et technologiques ont conduit à la mise au point de toute une gamme de procédés et d'équipements spécialisés qui permettent de trouver dans chaque cas la solution optimale.

Utilization of the LNG Cold Potential with Particular Reference to the Manufacture of Ethylene

by Ludwig Kniel*

1) Introduction

Ever since the advent of LNG and of the shipment of LNG across the seas, much thought has been given to the proposition of making these operations more profitable. There is a definite trend now to enlarge their scale so that unit costs may be brought down. Much larger LNG carriers are just a short while off and there is little doubt that larger capacity import schemes will take shape if, for instance, the present shrinkage in U.S. available gas reserves cannot soon be arrested by new discoveries.

Much painstaking work has been done by the people that have an interest in the Arzew-Le Havre-Canvey route and the experience gained with this and with projects presently under construction by others will help, one would hope, to make future schemes still more dependable and reduce operating costs.

One effort to reduce costs concerns the proposition to recover the cold potential inherent in liquified natural gas by virtue of its low temperature. The question is: cannot the revaporization process be coupled with other processes that require refrigeration such that the potential, i.e. the capacity to do useful work stored in the LNG, is recovered to the maximum possible extent and in a practical and cost saving manner?

The proposition of "back hauling nitrogen", which, I may assume you know about from the literature, has no commercial appeal. Nitrogen in such quantities would not be available. It must be produced, condensed against vaporizing LNG, stored, transhipped, stored again eventually vaporized to help condense some fraction of the natural gas only to have it escape to the atmosphere. Storage alone would require a huge amount of capital. It seems incongruous to think of hauling energy of such a tenuous nature as the reversible work contained in liquid nitrogen from the receiving terminal, where fuel costs are high, to the liquifaction terminal where fuel costs are low.

It has also been proposed that the LNG cold potential could be gainfully employed in air separation plants or for the cryogenic separation of hydrogen and, I might add of ethylene, from coke oven gases. We see little incentive also in these propositions. In either of these plants the required low temperature refrigeration potential is packed into the air or coke oven gas by compressing it before it enters the plant. The reversing exchangers or regenerators which do store the refrigeration also perform a second and equally important function, that of eliminating impurities in periodic cycles. They could not easily be replaced by refrigeration contained in LNG without fundamental changes to the process.

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By contrast the utilization of the LNG cold potential for the purification of ethylene involves a simple substitution of available LNG refrigeration for in-plant refrigeration. It can be applied today and without reservations.

It will become apparent later that this application too entails a substantial thermodynamic irreversibility, but it has the advantage of an organic integration with the primary process of revaporization. The feed stock to the ethylene plant, primarily ethane and propane, if present in sufficient volume, may be derived from the LNG in the course of revaporization to produce a specification grade send-out gas. If the LNG does not contain adequate amounts of these components to equal the tonnage required for an ethylene plant of optimum size, that will not impair the value of the combination. The LNG derived amounts of feedstock may in this case be supplemented with stocks from external sources within limits, which we hope to establish as this investigation progresses.

2) Energy Requirements for LNG Liquifaction

It is the reversible work of liquifaction that is available for recuperation during vaporization of the LNG at the receiving terminal and which, to abbreviate, we have called the cold potential.

For a single component the reversible work may be calculated from the well-known formula:

$$\boxed{W_{rev} = T\,(S_1 - S_0) - (i_1 - i_0)}$$

i = heat content; s = entropy

The subscript 1 refers to the initial, subscript 0 to the final (liquified) state and T means the absolute temperature, all in consistent units. With the simplifying assumption that the components behave ideally i.e. that they are completely soluble in one another and that there are no heat effects, the reversible work for mixtures, such as air or a natural gas, can be approximated by the same formula. Table 1 below gives comparative data for methane and two natural gases for an initial state of 750 psia (53 kg/cm^{2}abs) and 90°F (32.2°C).

Table 1

Reversible Work of Liquefaction .

Initial State: 750 psia, 90°F
Final State: Liquid at NBP (14.7 psia)

Type of Gas	Methane	Far Eastern	Lybian
Density (air = 1.0)	.554	.630	.789
Mol Weight	16.0	18.2	22.88
bhph/h per 10^6 scfd	149.7	167.4	177.9
bhph/h per 10^9 btu/d	148.0	147.4	135.0
btu/btu (LHV) to liquify	.0101	.0100	.0093
NBP °F	-259	-258	-256

The effect of pressure and density at NTP on the reversible work is shown graphically in Figure 1. You will note from Table 1 that the reversible work per unit of volume will increase as the density at NTP goes up, while per unit of its calorific value it will decrease which is of course due to the inclusion of fractions heavier than methane which have a considerably higher calorific value per unit of volume than has methane.

In caloric units the reversible work for the conditions of Table 1 will equal about one percent of the low heat value of the LNG. Thus the reversible work provides a measure by which to judge the efficiency of a liquifaction plant.

As an example, it is reported that the Arzew plant today burns about 18 cubic feet of gas per every 100 cubic feet liquified. The Alaskan plant, which will benefit from the icy cooling water of Cook's Inlet, might consume 12 to 14 cubic feet. The reversible work to liquify, from Table 1 will only require 1.0 cubic foot per 100, which corrected for the lower pressure of 450 psig at Arzew (Figure 1) will make 1.18 per 100 cubic feet. An efficient cascade with 80°F cooling water will have a cycle efficiency of 40 to 42% but that of the steam driven prime movers will hardly exceed 22 to 24% so that actual fuel consumption to liquify will be 1.18 x 1/.41 x 1/.23 = 12.5%. The greater part of the remaining difference up to 18% may be attributed to services. Higher efficiencies could of course be attained at extra expense, but where fuel is ample and inexpensive high thermal efficiency need not be an objective while that of minimum initial cost will definitely be one.

3) Revaporization of LNG

As in the case of liquifaction the reversible work recoverable on vaporization will identically depend on the pressure which will now be the send-out pressure. The send-out pressure will be set by the utility which takes care of distribution, in accordance with its requirements.

What portion of the reversible work will actually be recovered will depend on how the process of vaporization is conducted. It will particularly depend on how closely it is possible to approach the "vaporization curve", better identified as the heat content vs temperature curve of the LNG with the heating fluid of the complimentary process, in our case the ethylene bearing gas of an ethylene plant. Figure 2 shows the heat content vs temperature curve for the Lybian gas of Table 1 at 450 and 600 psia pressure.

Consider now a plant for a "normal" vaporization rate of 125 million cubic feet per day, which by definition shall be equivalent of one day's receipts averaged over the calendar year and for a 600 psig send-out pressure. If the LNG is practically all methane, as is the Alaskan gas, the maximum recoverable work will be .01085 btu per btu (LHV) which is equivalent to 15 000 Kwh/h at a flow of 125 million scfd. For the Lybian gas of

Table 1 which will increasingly engage our attention, the corresponding figure is 16 600 kwh/h. These figures pertain to complete vaporization, the vaporized gas leaving at 60°F and 600 psig pressure.

In the case of methane the vaporization facility would consist of a series of exchangers acting as chillers for the raw ethylene gas.

The Lybian LNG however contains substantial amounts of ethane, propane etc and requires fractionation to obtain a send out gas of specified heat content. It will also be profitable then to recover ethane, propane etc separately for disposal as existing market conditions might warrant.

The left half of Figure 3 shows schematically a LNG fractionation and vaporization unit tailored to the Lybian gas composition. To describe it very briefly, LNG from atmospheric storage will be pumped over a series of vaporizing exchangers, summarily called the "cold exchanger train" and will enter the demethanizer partly vaporized. The overhead vapor stream from the demethanizer constitutes the send out gas which will be accurately controlled as to its ethane content which means its calorific value. It will subsequently be superheated to the send-out temperature.

The heavier fractions after methane will successively enter the deethanizer for removal of all of the ethane, the depropanizer for removal of a propane - LPG fraction and perhaps even a debutanizer (not shown).

The ethane will nearly all be withdrawn in the vapor state, only a small amount going to liquid storage which is located within the ethylene plant. Propane and all heavier fractions will leave as liquid products.

The material balance in Table 2 below has been drawn up for a flow of 125 million standard cubic feet per day of the Lybian gas of Table 1, a send-out gas calorific value of 1020 to 1025 btu (HHV) per cubic foot (9595 to 9645 kcal per Nm^3) and describes in detail the first two separations. There would be produced from the LNG deethanizer 23.25 metric tons/hr of ethane which in 355 operating days - granting for the moment that the ethylene plant can be operated this length of time per year - will yield 152000 mta (metric tons per annum) of high purity ethylene. If all of the available propane would also be charged to the ethylene plant, the ethylene production would mount to 262000 mta, a respectable output even in these days of huge plants.

Considering now the available cold potential, it will be obvious that firstly the fractions removed in the liquid phase will not contribute their full share to what would be recoverable if the LNG were totally vaporized to leave at 60°F. Secondly the demethanizer and deethanizer of the fractionation system demand a certain amount of

Table 2

LNG MATERIAL BALANCE

Capacity: 125 million scfd Lybian LNG

Component	LNG Feed Vol %	LNG Feed lbmols/hr	Demethanizer Overhead = Send out gas lbmols/hr	Deethanizer Overhead lbmols/hr	Dethanizer Bottoms lbmols/hr
N_2	0.9	123.7	123.7		
CH_4	70.0	9 619.7	9 585.7	34.0	
C_2H_6	15.0	2 061.3	356.3	1 675.0	30.0
C_3H_8	10.0	1 374.2		8.0	1 366.2
Iso-C_4H_{10}	1.4	192.4			192.4
No-C_4H_{10}	2.1	288.6			288.6
C_5H_{12}	.5	68.7			68.7
C_6H_{14} plus	.1	13.7			13.7
Total	100.0	13 742.3	10 065.7	1 717.0	1 959.6
Thousand scfd		125 000	91 600	15 600	17 800
lbs/hr		314 426	167 934	51 265	95 227
Kg/hr		142 597	76 161	23 249	43 187
Mol weight		22.88	16.68	29.86	48.60
Density (air = 1.0)		.789	.575		1.677
HHV, btu/scf$_3$		1 365	1 024	1 760	2 760
HHV, Kcal/Nm		12 830	9 640	16 550.	25 960
Total 10^3 Therms/day		1 704	938	275	492
Total 10^3 Toncals/day		43 000	23 650	6 930	12 410

refrigeration at specifiable temperature levels for their operation and this demand must be met with LNG (there being no other source) which will further reduce the available cold potential.

These reductions can accurately be calculated for any gas composition and any pressure. I need not tire you with details; suffice it that I say that the degree of vaporization of the demethanizer feed turns out to be an important parameter.

For the Lybian gas the optimum feed temperature at 600 psig appears close to -30°F where 58 vol % of the feed are vaporized. A gain would be had if this temperature could be higher. However, the feed would then contain increasing amounts of ethane in the vapor phase which would call for more reflux to be produced with refrigeration at -105°F. The gain would thus be more than nullified. The reverse holds true for feed temperatures below the optimum. Figure 4 shows the condition of the feed at temperatures higher than the bubble point for the Lybian gas mixture.

After making the aforementioned reductions the heat content versus temperature curve assumes the shape in Figure 5. You will notice that the corrections affect primarily the high temperature range down to about -10°F as one would expect from the lack of heavy fractions in the send-out gas. The effect is less pronounced in the medium temperature range to about -140°F. At still lower temperatures the heat content curve remains unchanged.

4) Refrigeration Requirements of Ethylene Plants

As introduction to a discussion of refrigeration requirements of ethylene plants it will be helpful to refer briefly again to Figure 2 of which the right hand half conveys an idea of the principal elements of such a plant. The important item here is again the "cold exchanger train." It cools the ethylene bearing gas stream, after it has been compressed, to a temperature at which practically all of the ethylene and most of the methane are liquified. An impure hydrogen stream may be removed at this point. The refrigeration levels will extend down to about -150°F and plant refrigeration is normally furnished by an ethylene-propylene cascade.

In most plants compression of the raw gas is carried to a pressure of 500 plus psig (35 Kg/cm^2 ga.) At the corresponding somewhat lower pressure of 475 psig at the outlet end of the cold train, condensation of the ethylene will be reasonably complete near the -150°F level, i.e. at a temperature within reach of atmospheric ethylene as the refrigerant. A methane refrigeration stage will thus not be needed.

With low temperature LNG as the refrigerant, compression need only be carried to about 90 psig i.e. to a pressure such that the fuel gas components, which return over the cold train exchangers, will be at 30 to 40 psig and will not require recompression. This pressure reduction will make the charge gas compressor less than one half the size of that in a conventional plant. The refrigeration plant however cannot be entirely deleted.

The ethylene column requires the removal of a rather large amount of heat at about the -20°F level to produce reflux. The heat removed can be absorbed by the reboiler. In other words the heat balance of the ethylene column will best be served by a heat pump operating between temperature levels some 50°F apart. In the case where the ethylene is removed as a vapor product the column heat balance will nearly be met by the heat pump cycle but an allowance must be made for some amount of refrigeration from the LNG side to trim the operation of the ethylene column. To have the refrigeration requirements of the ethylene column provided entirely from the LNG side would introduce complications and would make the combination less attractive.

A comparison of the compressor requirements of a conventional plant producing 152 000 mta of high purity ethylene from ethane versus one served by an LNG refrigerant as explained, shows the following:

Table 3

Comparative Compressor Requirements

Capacity: 152 000 mta of Ethylene

		Conventional	LNG Served
Charge Compressor,	bhp	10 900	4 650
Refrigeration,	bhp	8 700	2 300
Total,	bhp	19 700	6 950
Total,	Kw	14 700	5 190

The cooling requirements for a 152 000 mta ethylene from ethane plant, excepting those handled by the heat pump, but taking credit for refrigeration recuperated from returning product streams are shown graphically in Figure 6. To illustrate the dependence on pressure, curves for the conventional as well as the LNG connected cold train were plotted on this graph.

It is only necessary now to transfer these curves on to the plot of Figure 5 to obtain a graphic picture of the efficiency of utilization of the LNG cold potential in combination with an ethylene plant, see Figure 7. This plot yields some interesting results.

As the first of these we note that the cold potential contained in 125 million scfd of the Lybian LNG would be adequate for the needs of about a 325 000 mta ethylene plant using ethane and propane as charge stocks.

Secondly, a 152 000 mta plant which would obtain its feed stock solely as ethane from the Lybian LNG provides a wholly inadequate complement. It would furnish a scanty third of the heat necessary to vaporize the LNG over the LPG fractionation system. Two thirds would go to cool down the ocean. Probably close to 90% of the potential or reversible work would be dissipated.

Thirdly, to obtain reasonable utilization of the cold potential ethylene capacity should be in the order of at least twice the amount that can be produced from ethane contained in the LNG, or about 300 000 plus tons per year.

What then would be the benefits to the ethylene operation from such a merger? The reply to this question is that these are nevertheless considerable.` For one thing there is capital cost. The capital cost of a 152 000 mta from ethane conventional plant will be 14.0 million dollars, that of the plant served by LNG refrigeration only about 10.6 million dollars.

The following Table 4 gives a comparison of production costs for an ethane cost of 45 cents per million bru (HHV) equivalent to 1.0 cent per pound.

Table 4

Comparative Production Costs

Capacity: 152 000 mta of Ethylene

	Conventional $/mt	LNG Combined $/mt
1) Charge Stock (Ethane 1.0 cents/lb)	28.9	28.9
2) Total Operating Expenses (Fixed and Other)	25.6	17.0
3) Byproduct Credit	(8.8)	(11.7)*
4) Production Cost	45.7	34.2
5) Return on Investment before taxes (25%/year of 1.2 times cost of Ethylene Plant)	27.6	21.0
6) Total Cost of Ethylene		
$/mt of Ethylene	73.3	55.2
c/lb of Ethylene	3.32	2.50

* Includes credit for surplus steam (exported at $1.20 per 1 000 lbs.) available because of lower compression energy requirements.

The figures of Table 4 do indicate that the value of LNG refrigeration to an ethylene plant of the size referred to and designed as described will be of the order of eight-tenths cents per pound of ethylene and that the ethylene manufacturing cost may approximate 2 1/2 cents per pound. How this credit,which will be of the order of 2.7 million dollars per year,should be divided must be agreed upon between operators of the LNG terminal and of the ethylene plant. If, in the light of what has been said before, the ethylene plant capacity could be doubled the credit would reach the substantial sum of 5.4 million dollars per year and the manufacturing cost per pound would be reduced further due to the effect of size on the necessary investment.

5) Coupling an Ethylene Plant to an LNV Vaporization Facility

It has thus far been tacitly assumed that the LNG feed rate would be constant and that as a consequence the ethane recovery and ethylene production rates would also remain constant.

But the demand for send out gas will vary daily and seasonally. During these variations the ethylene production rate should remain essentially constant not only because the plant will operate most profitably at constant rate but also with respect to furnishing ethylene at constant rate to any derivatives plants.

As an example, the utility that takes possession of the send out gas may require a "turn down" ratio referred to normal capacity of .5 and a maximum send out rate of 1.3 times normal. Such a situation, as we have seen, could be met with the Lybian gas. A 125 million scfd vaporization facility could be turned down to half the normal rate and still furnish ample refrigeration for a 152 000 mta ethylene plant. The ethane feed stock during times of oversupply would have to be stored and drawn from storage while the send out rate drops below normal. With this arrangement cold utilization will remain poor due to inadequate capacity at the ethylene plant.

To improve utilization it will be necessary to firstly enlarge the ethylene plant and to provide additional feedstock in the form of the LNG contained propane and butanes or provide such stocks from outside sources. Secondly and more importantly it will be necessary for the utility to fix the turndown ratio at a higher figure by inducing their "around the clock" industrial consumers to take more of their base load with gas and less with fuel oil. This should not be too difficult a task considering the advantages of gas as an industrial fuel and the relative price differential between perhaps 55 to 60 cents per million btu (LHV) for gas and 42 cents (2.50 $/Bbl) for low sulfur fuel oil.

Such an enlarged ethylene plant could be operated with a predetermined ratio of feeds namely the amount of ethane available at the lowest sent out rate, say at .80 times the normal, plus whatever supplementary feed stock is selected to bring the plant to capacity, which cannot be more than corresponds to the refrigeration potential available at the lowest sent-out rate. At higher sent out rates ethane would then be in oversupply. It could be disposed of currently by blending it with hydrogen rich ethylene plant off gases for the same calorific value, nearly the same density and a Wobbe number W' well within less than 3% of the primary send-out gas. In a manner of speaking, one would peak-shave with an ethane-hydrogen blend. The hydrogen content of the total send out gas at 1.4 times the normal send-out rate would remain below 4.0 vol%.

6) Conclusions

Use of the LNG cold potential for the purification of ethylene will result in substantially lower cost ethylene from a medium sized plant.

As long as fractionation of the received LNG is required to produce a proper heat value send-out gas, the recovery of ethane and propane from the LNG will be worthwhile. It is evident however that the quantities so recovered will not ever be adequate to make commensurate use of the LNG potential in the purification of the resultant ethylene. Additional feed stock must be procured from external sources. This may be additional ethane or propane or even butanes and natural gasoline. Treatment of the latter two for ethylene requires more refrigeration per pound of ethylene but such additional refrigeration will all be at high temperature levels where an ample supply still remains untapped as shown in Figure 7.

More complete recuperation will also require that the turn-down ratio of the gas send-out rate be no lower than say .8 times normal. Peak loads do not harm such a modus of operation and peak shaving with the hydrogen rich off-gas from the ethylene plant appears an interesting possibility.

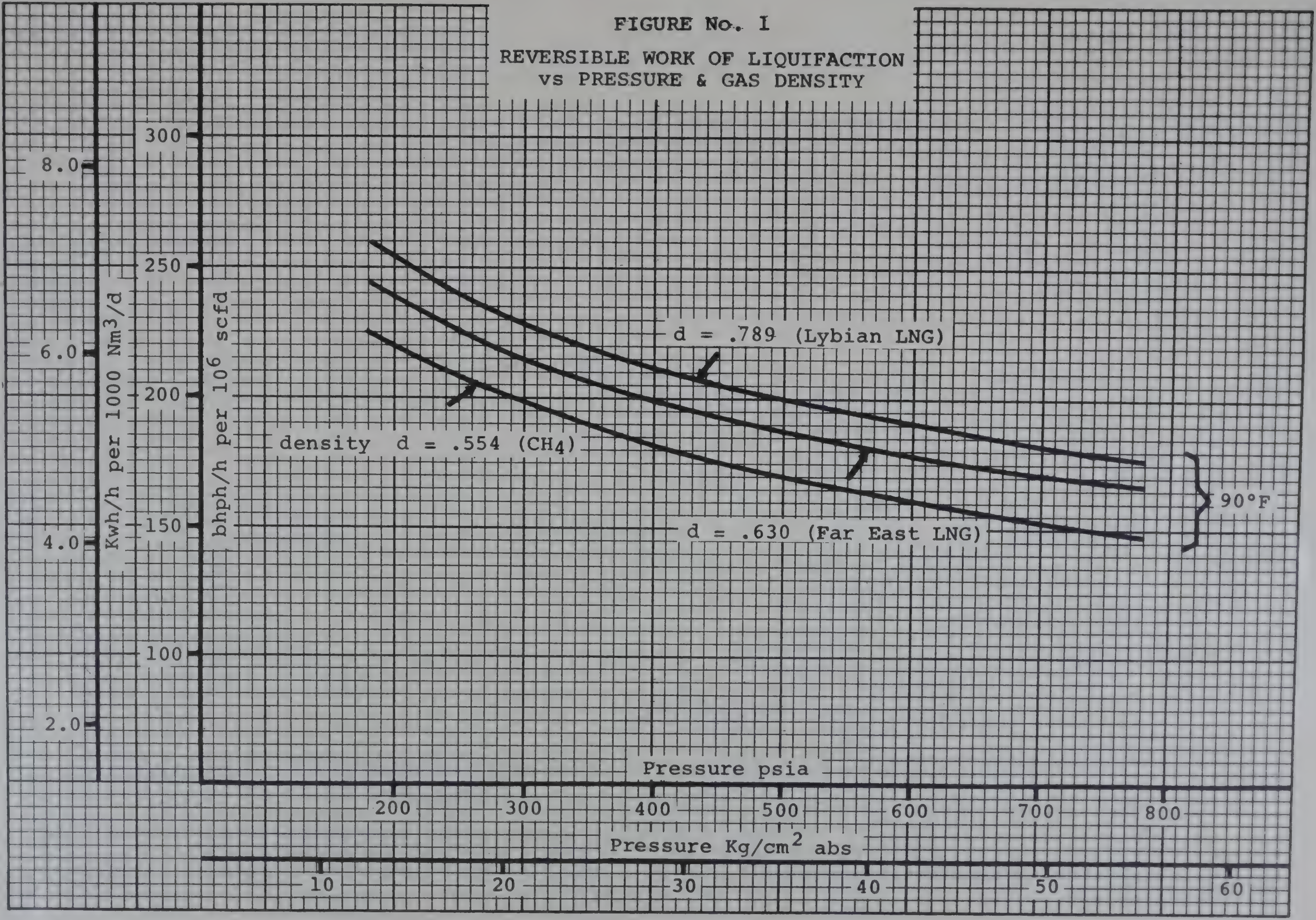
Lowest cost ethylene could be produced from such a combination of LNG facility and ethylene plant. The realization of such a project however will depend on full cooperation between the terminal authority, the gas utility and the ethylene plant operators.

7) Acknowledgement

The valuable assistance of Mr. K. Farid, Lummus Process Engineer, with analysis and computer calculations is gratefully acknowledged.

FIGURE No. I

REVERSIBLE WORK OF LIQUIFACTION
vs PRESSURE & GAS DENSITY



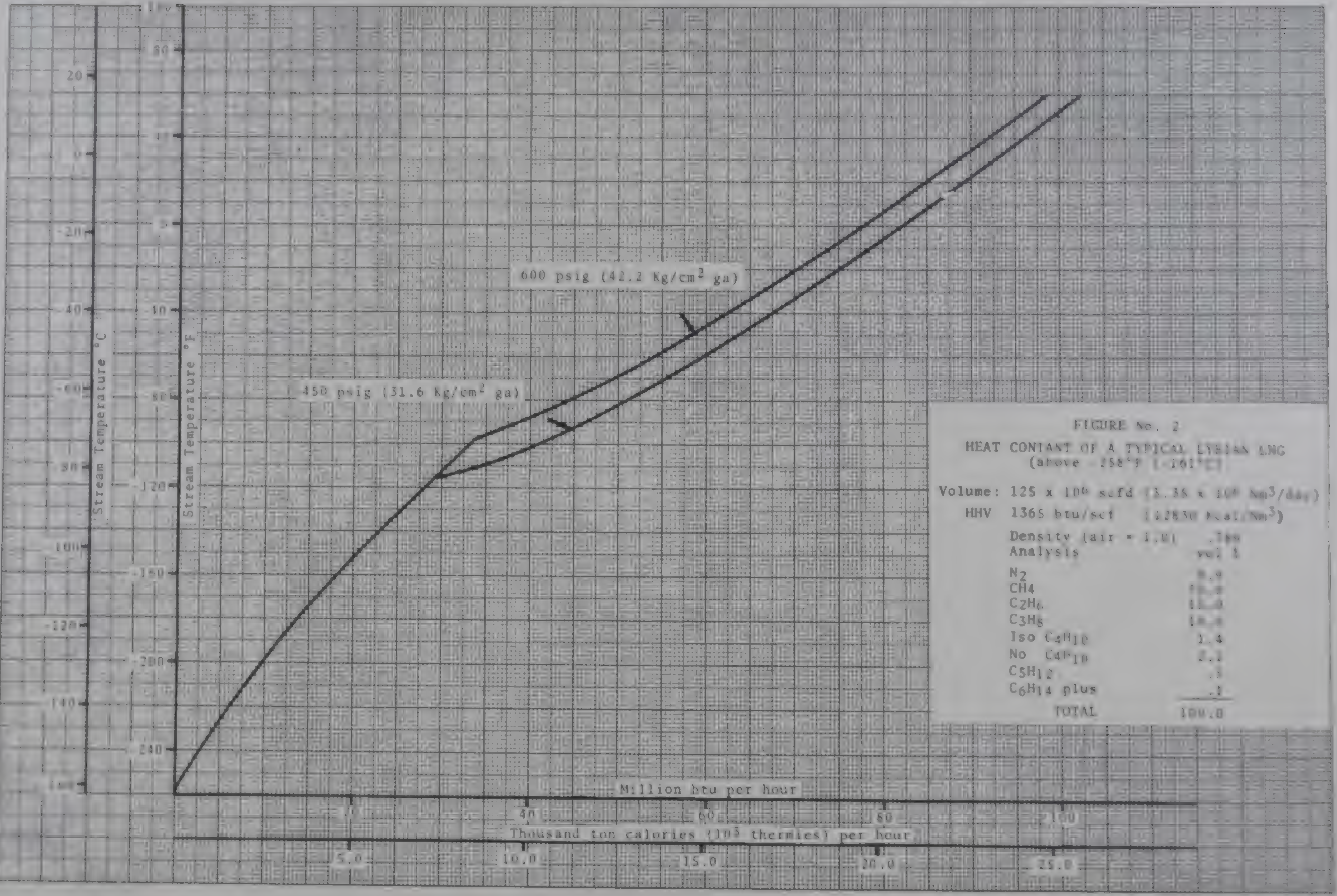


FIGURE No. 2

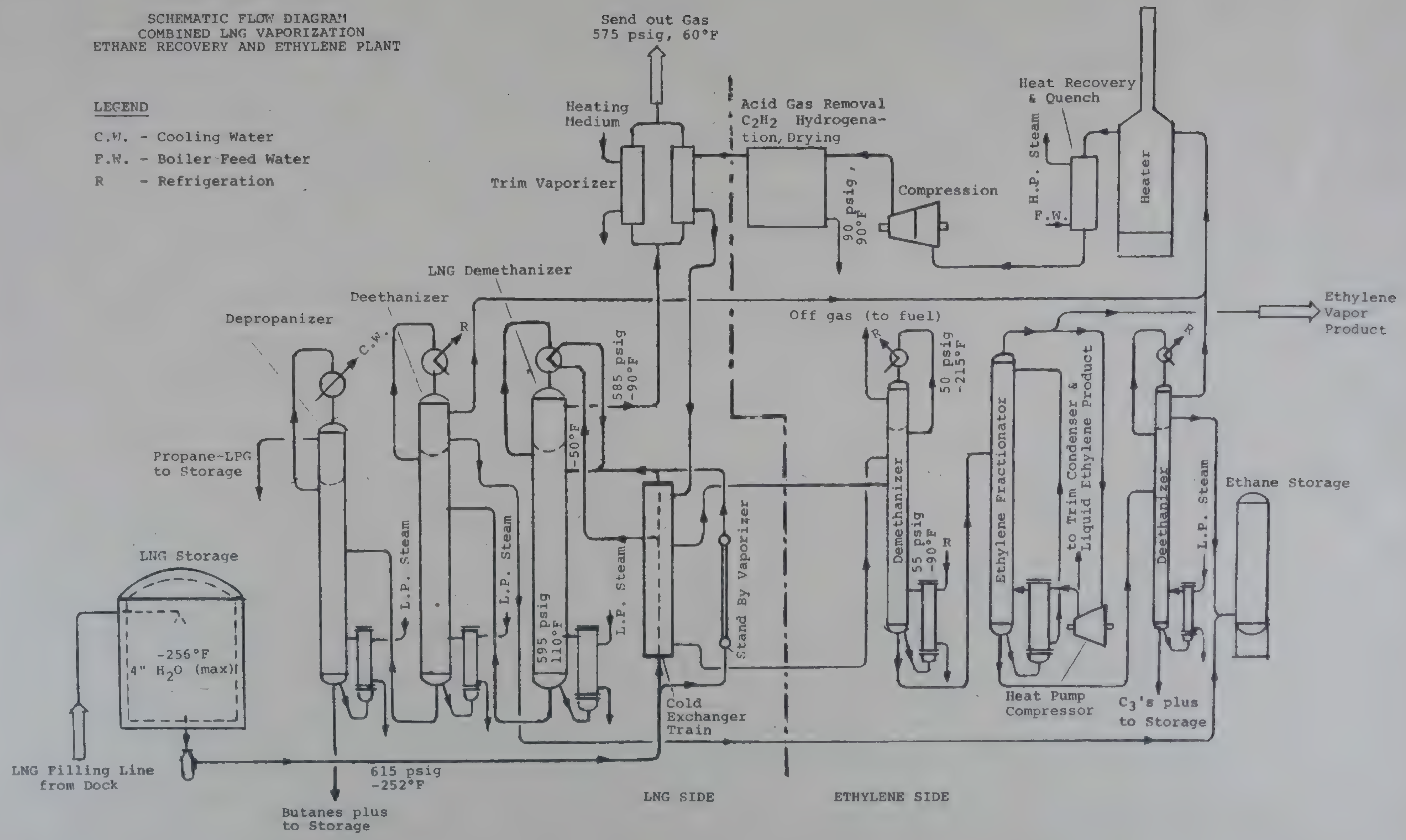
HEAT CONTANT OF A TYPICAL LYBIAN LNG
(above -258°F [-161°C])

Volume: 125 x 10^6 scfd (3.35 x 10^6 Nm^3/day)
HHV 1365 btu/scf (12830 kcal/Nm^3)

Density (air = 1.0)	[illegible]
Analysis	vol %
N_2	0.9
CH_4	[illegible]
C_2H_6	[illegible]
C_3H_8	[illegible]
Iso C_4H_{10}	1.4
No C_4H_{10}	2.1
C_5H_{12}	.3
C_6H_{14} plus	.1
TOTAL	100.0

FIGURE No. 3

SCHEMATIC FLOW DIAGRAM
COMBINED LNG VAPORIZATION
ETHANE RECOVERY AND ETHYLENE PLANT


LEGEND
C.W. - Cooling Water
F.W. - Boiler Feed Water
R - Refrigeration
Send out Gas
575 psig, 60°F
Heating Medium
Trim Vaporizer
Acid Gas Removal
C_2H_2 Hydrogenation, Drying
90 psig, 90°F
Compression
Heat Recovery & Quench
H.P. Steam
F.W.
Heater
LNG Demethanizer
Deethanizer
Depropanizer
C.W.
R
585 psig
-90°F
-50°F
595 psig
110°F
L.P. Steam
L.P. Steam
L.P. Steam
Propane-LPG to Storage
LNG Storage
-256°F
4" H_2O (max)
LNG Filling Line from Dock
615 psig
-252°F
Butanes plus to Storage
Cold Exchanger Train
Stand By Vaporizer
LNG SIDE
ETHYLENE SIDE
Off gas (to fuel)
R
50 psig
-215°F
Demethanizer
55 psig
-90°F
R
Ethylene Fractionator
to Trim Condenser & Liquid Ethylene Product
Heat Pump Compressor
R
Deethanizer
L.P. Steam
C_3's plus to Storage
Ethane Storage
Ethylene Vapor Product

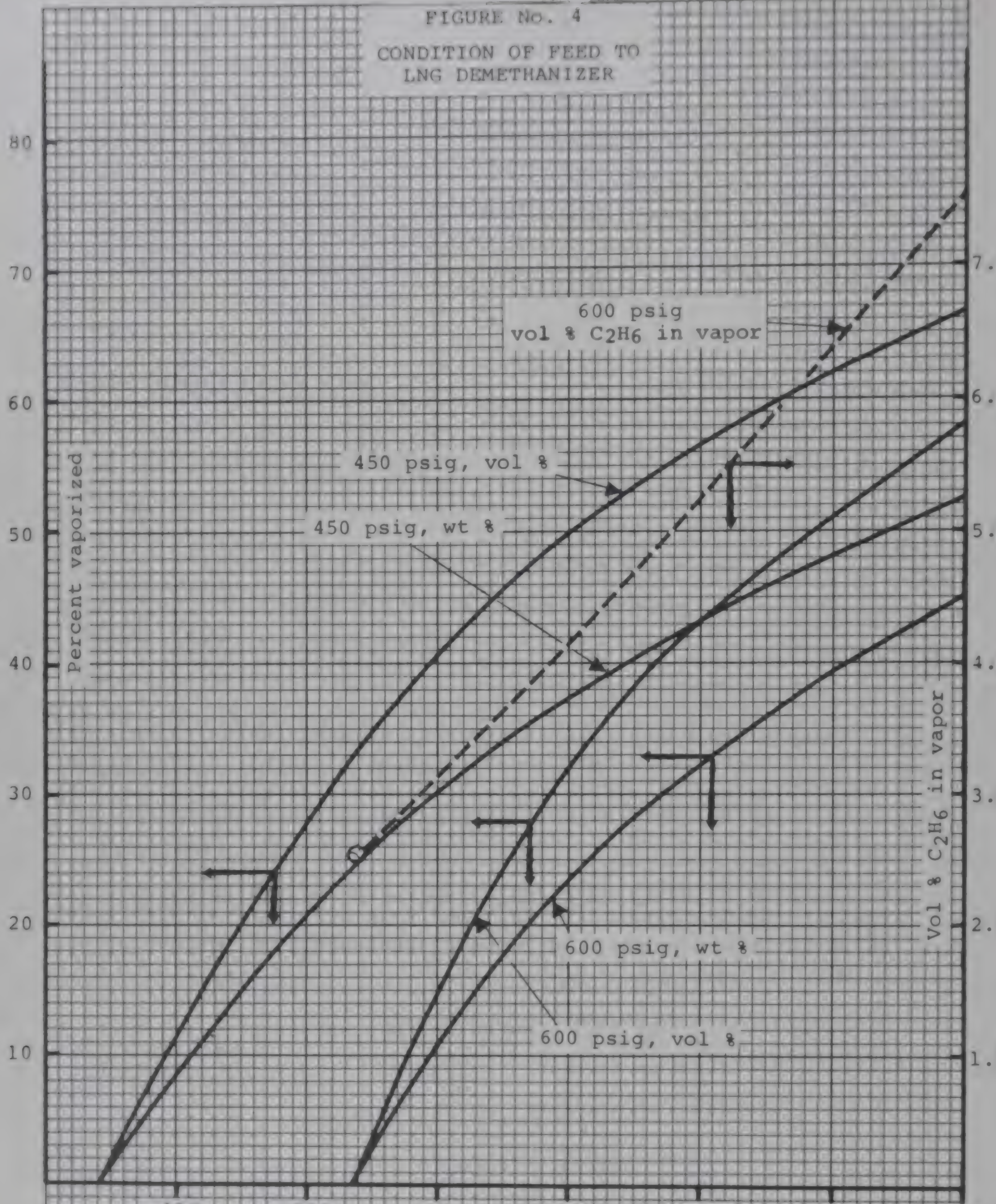


FIGURE No. 4

CONDITION OF FEED TO LNG DEMETHANIZER

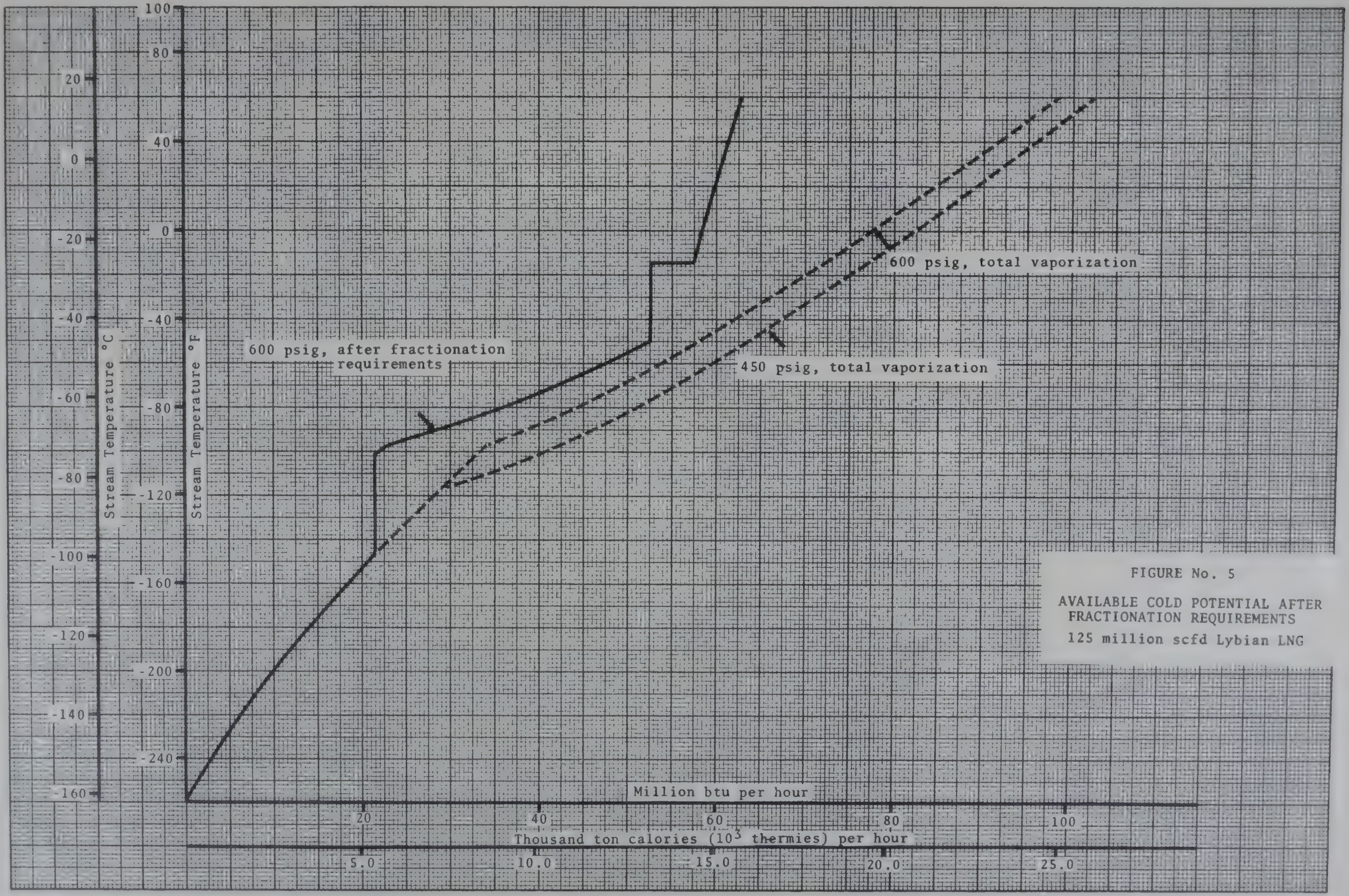


FIGURE No. 5

AVAILABLE COLD POTENTIAL AFTER FRACTIONATION REQUIREMENTS

125 million scfd Lybian LNG

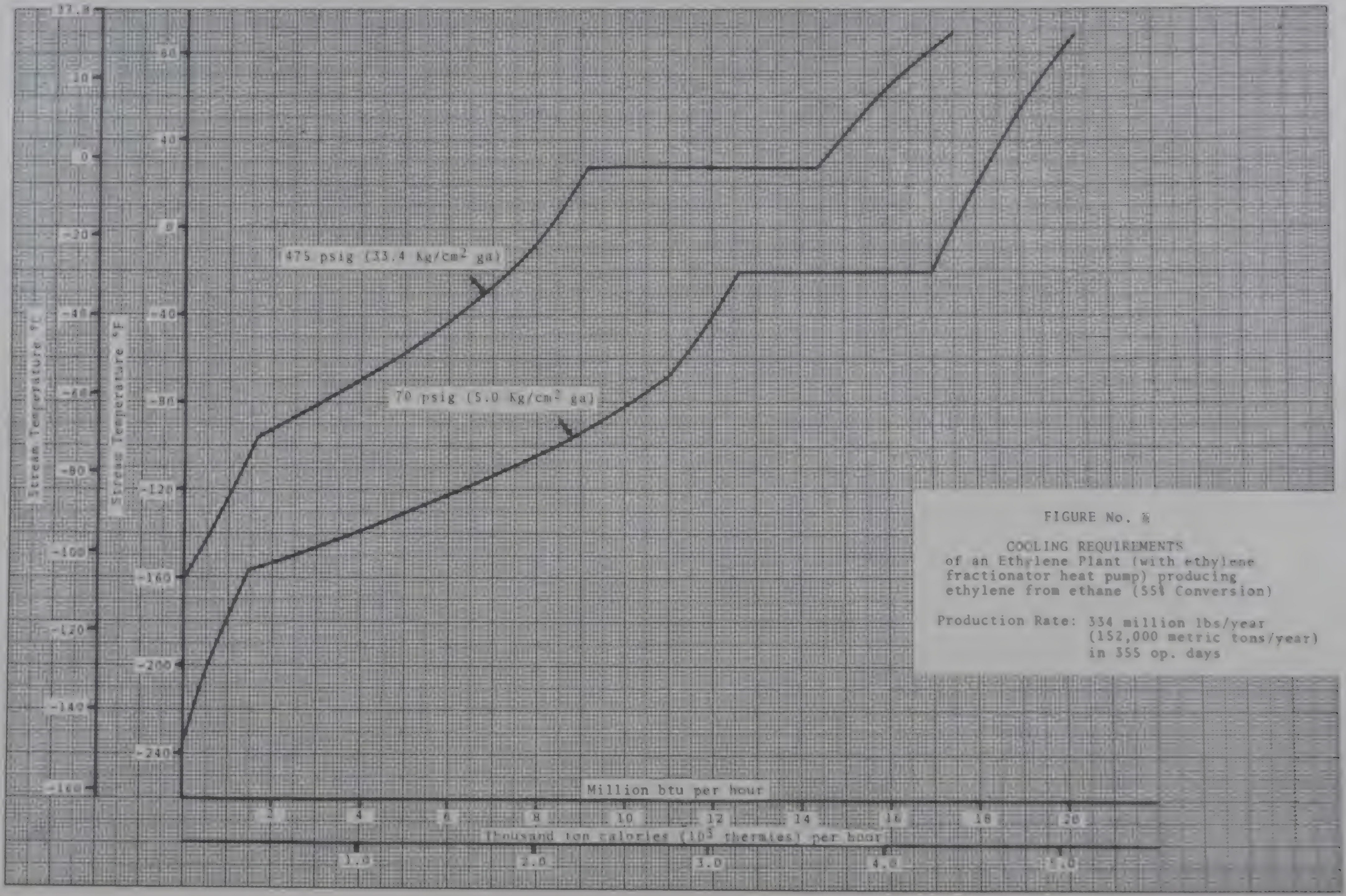


FIGURE No. 8

COOLING REQUIREMENTS
of an Ethylene Plant (with ethylene fractionator heat pump) producing ethylene from ethane (55% Conversion)

Production Rate: 334 million lbs/year
(152,000 metric tons/year)
in 355 op. days

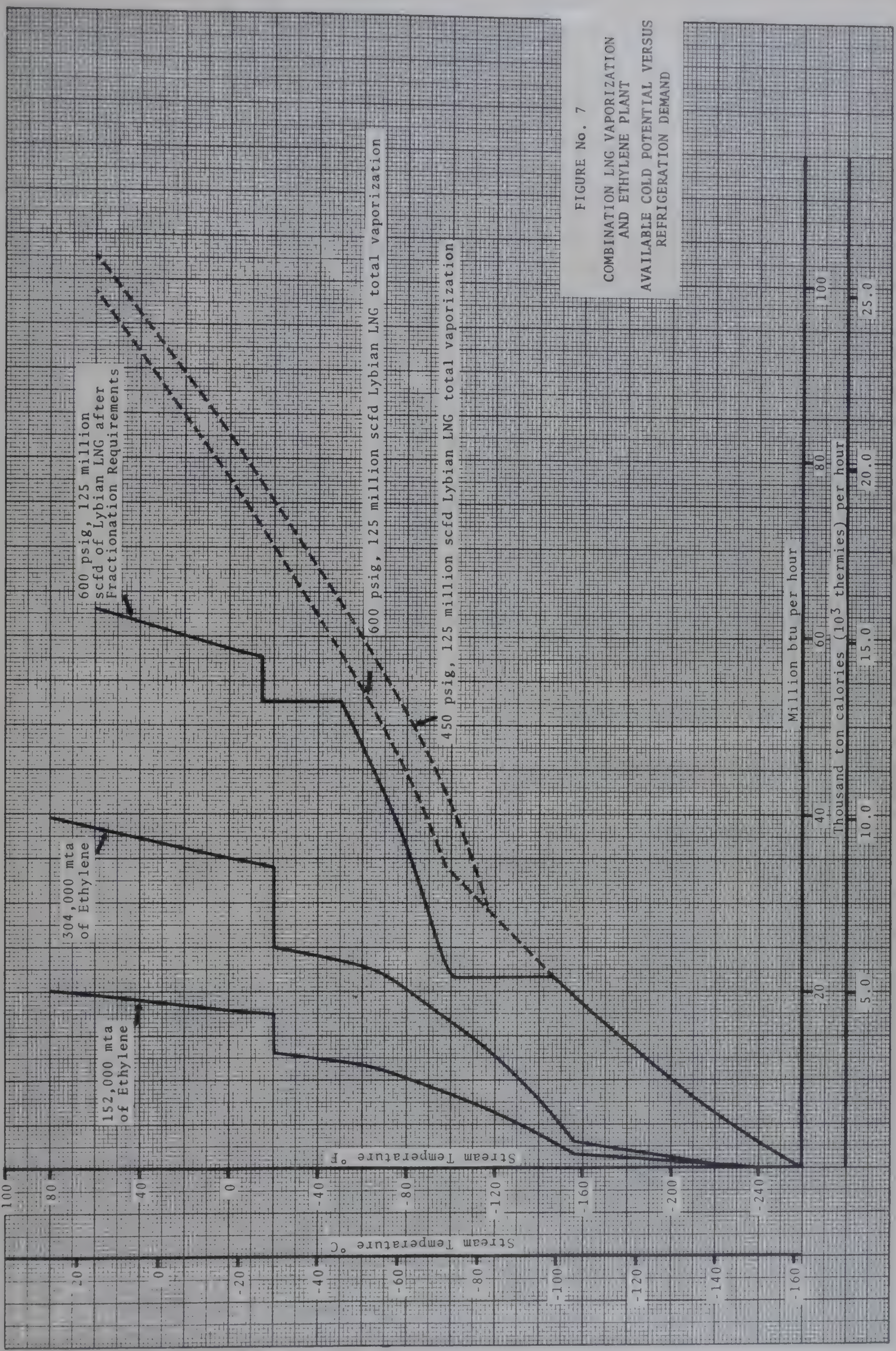


FIGURE No. 7

COMBINATION LNG VAPORIZATION AND ETHYLENE PLANT

AVAILABLE COLD POTENTIAL VERSUS REFRIGERATION DEMAND

SOME ASPECTS OF THE UTILIZATION OF LNG FOR THE PRODUCTION OF POWER IN I.C. ENGINES

By

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ABSTRACT

Review is made of the main technical aspects of the utilization of natural gas as the primary fuel in internal combustion engines of both the piston and gas turbine types for the production of power. This review is then extended to consider the further problems brought forward by the utilization of LNG in conventional internal combustion power plants.

Many of the technical problems arising from the applications of LNG in engines are similar to those when NG is the fuel. Much research and experience in this area exist. There is, however, very much less information regarding the further problems brought about by the relatively high heat sink of LNG when used as an engine fuel. Expansion in this field of application of LNG is very much dependent on the further technological advances needed for the handling of a cryogenic fluid in everyday applications.

INTRODUCTION

Natural gas has many virtues as an engine fuel. In addition to being cheap and abundant, it is of a high heating value on weight basis, and a relatively clean gas. Nonetheless, it has some drawbacks in that it has a large volume per unit of heating value, it is not readily transportable and difficult to store at the point of use. If natural gas were to be liquified however, it will occupy only around one six hundredth its volume as a gas and its transport to areas not accessible by pipelines becomes feasible. In recent years considerable progress has been made in the technology of the manufacture, transportation, storage and regasification of liquid natural gas by a rapidly growing industry.

Means for the utilization of natural gas for generating power whether in large or small capacity units are well established and it is only a matter of consumer's choice to decide on the type of equipment to be used for the job. Moreover, some limited experience exists with the use of LNG for the production of power. There is however, much room for research and further experience in overcoming the problems which will arise in the widespread application of LNG in such devices.

Natural gas burning engines, are available in both the reciprocating type piston engines, and the rotating type, mainly gas turbines. The older type of piston engines ran on mixtures not far removed from the stoichiometric mixture and employed low compression ratios. A too high compression ratio in these engines would have produced severe knocking, preignition, loss of power, and possible engine damage. Consequently, they were seriously inefficient. Knowing that higher engine compression reduces fuel consumption and operating costs, means have long been sought to burn gaseous fuels at higher compression ratios. It is only relatively recently, that engines have been developed to burn gaseous fuels using compression ratios similar to those of diesel engines and thus achieve similar efficiencies. This has been largely possible by means of operating on learner mixtures and proper attention to combustion chamber design.

Today's gas turbines operating on natural gas produce outputs of more than four times those of 1939 for the same frame size. Such an improvement in performance has been possible mainly by proper attention to the design of the plant components and by the development of new materials capable of withstanding high temperatures and better cooling techniques. When economy is the main concern, the gas piston engine is more attractive than gas turbines. However, where reliability and availability are of more importance, as in emergency power applications, gas turbines would be prefered. Gas turbines are attractive because of their relative simplicity, ease of maintenance, the longer life they have and the higher power weight ratio.

The present paper reviews the nature of the problems normally associated with the use of natural gas in internal combustion plants in general, and reciprocating machinery in particular, for the production of power. This review is then extended to consider the further problems brought forward by the utilization of liquified natural gas in conventional internal combustion power plants and some suggestions are made for efficient modifications to make such applications more competitive with other fuels including natural gas.

NATURAL GAS AS A PRIMARY FUEL IN PISTON ENGINES

There are two main types of gas piston engines available in high compression. These are the dual fuel engine and the spark ignition engine. The first of these engines can be operated on either a mixture of a small quantity pilot fuel oil and gas, which in most cases constitutes the larger percentage of fuel, or fuel oil alone. The spark ignited gas engine, can only be operated on gaseous fuel and thus its application is only suitable where the gas supply is adequate and uninterrupted.

The efficiency of the dual fuel engine in terms of the heating value of the gas and the pilot oil consumed per unit of power output is approximately the same as when running as a diesel. However, these engines are noted to possess two main limitations associated with the poor utilization of the gaseous fuel at light load and the loss of combustion control commonly termed as the onset of knock which impose practical limits to the useful output of these engines.

It is well established that in a dual fuel engine the poor part load performance is due mainly to an appreciable proportion of the fuel gas surviving the combustion process when fed to the engine at well below a limiting concentration, (1). This limit is identified with the effective flammability limit of the mixture corresponding to engine conditions. Some improvement of part load performance can be achieved by partial air restriction at part load to result in a richer effective mixture. The extent of air throttling, however, must be controlled to avoid excessive reduction of the density after compression which may lead to irregular ignition. Moreover, preheating the charge and the employment of larger pilot quantities have appreciable effects on the extent of utilization of natural gas. On the other hand, the knock phenomena observed in the dual fuel engine is of an autaiguition nature, most probably of the gaseous mixture in the neighborhood of the ignition centres. The processes determining the onset of knock are essentially similar to those under motored or spark fired conditions. Therefore, the main factors influencing the occurrence of knock are the nature of the gaseous fuel, the charge preheat and to a lesser extent the pilot quantity. The control of knock has assumed particular significance recently with the trend of ever increasing outputs through supercharging.

Mixture control is perhaps even more important in spark ignited gas engines. The need for the mixture ratio in these engines to be held within close limits, is imposed by the high tendency to knock over the rich mixture range and by the increased cyclic irregularity producing erratic combustion, rough running, and high fuel consumption at the lean mixtures range. However, in terms of highest knock-free compression ratio, the knock rating of methane is 15.6 as compared with about 6 for hydrogen and 7.9 for propane, (2). As a result, piston engines running on natural gas can employ higher compression ratios and consequently have higher efficiencies and greater outputs than with other fuels. The operation of spark igntion engines on natural gas rather than a liquid fuel, such as petrol, indicates that for optimum power the slower burning rate of methane requires greater spark advance and narrower mixture strength range. Moreover, being a gaseous fuel the evaporative cooling of the charge is normally absent, hence producing a lower volumetric efficiency on air basis and operation tends to be more temperature sensitive, (3). Operation of spark ignited engines on natural gas indicates that exhaust emissions as far as hydrocarbons, carbon monoxide, and oxides of nitrogen are significantly reduced. Moreover, the hydrocarbons emmitted have normally a lower reactivity.

Figure 1 presents typical comparative performance curves for spark ignited and dual fuel versions of turbo-charged engines built to operate as a spark ignition, dual fuel or compression-igntion unit, (4). The dual fuel version shows an improvement in performance over that of the spark-ignited version throughout the load range considered. This improvement comes principally from the better combustion stability with dual fuel engine operation and its ability to operate at higher excess air ratios. The economics are usually, however, considered to be in favour of the spark-ignited version because of the relatively high cost of the few percent of the pilot required. The substantial reduction in crankcase oil contamination, the less frequent changes and cleaning of sparking plugs and the improved mixture distribution in multi-cylinder engines are added advantages when using natural gas as a fuel.

NATURAL GAS AS A PRIMARY FUEL IN GAS TURBINES

The basic simple gas turbine power plant has a relatively low thermal efficiency compared to the reciprocating type engine. There are, of course, several ways in which the thermal efficiency of the simple cycle may be improved, however with increased complexity and cost. The efficiency and specific power output of a simple gas turbine plant are particularly dependent on the compression pressure ratio and turbine inlet temperature used. Small gas turbines normally have a slightly lower efficiency than larger units. The life of the major components of any installation is much influenced by the values of the inlet gas temperature to the turbine and the frequency of starts from cold. The higher these values, the shorter will be the expected life of the installation. Proper selection of the operating inlet gas temperature is normally a compromise between the requirement for improved efficiency and reduced operating life.

Much experience has been accumulated by most of the large gas turbine manufacturers with operation on natural gas in stationary ground installations used for pumping duties or for the generation of electricity. Gas turbine units converted to run on natural gas are usually those which have been designed basically to burn liquid fuel. In most cases, these units need only little modification when they are switched from one fuel to another.

If the gas turbine were to be built with the specific intention of running on natural gas, design simplifications can be made as far as fuel injection, fuel atomization and mixing. The combustion is always better for fuels admitted in a gaseous rather than a liquid state. This is partly because of the somewhat hotter flame which results, and chiefly because the period for droplets to vaporize is elminated. This period, otherwise, may be of the same order as the residence time of the fuel in the combustion chamber. As a result the gas turbine combustion chamber, when natural gas is used as a fuel, can be of a more compact design and shorter length than those designed for liquid fuel operation.

As far as maintenance, dependability, and availability are concerned, operating experience with gas turbines has indicated that the units burning natural gas tend to score more heavily over machines burning other types of fuel, (5). Such a conclusion could be readily drawn from the fact that the burning of natural gas usually results in a minimum of smoke and carbon deposits, low flame luminosity, better cooling and hardly any corrosive residues. As a result deposits which usually accumulate on fuel burners, turbine blades and on the walls of heat exchanger tubes are appreciably reduced when burning natural gas. Such deposits when operating on liquid fuels, normally result in appreciably pressure drops, variation in performance and under severe conditions a loss of power of up to 25 per cent, (6). In addition, in some cases, the formation of these deposits aided by the formation of sulphur compounds, can lead to a severe corrosive attack on turbine blades, thus limiting their useful life.

LNG AS AN ENGINE FUEL

The ever increasing availability of natural gas from various sources throughout the world and recent advances in cryogenic science and technology, have brought about a rapidly emerging LNG industry. This has helped in making the transportation of natural gas in a liquified form in many cases economically feasible. A notable example is the bulk transportation of LNG from Algeria to the United Kingdom. Much attention has been focused recently on the problems and advantages of LNG as an engine fuel. This has been prompted largely by the accumulating experience in the handling of LNG and the continuing reduction in its cost together with an increasing pressure for air pollution control from combustion systems and motor vehicles in an expanding market for thermal power equipment.

The advantages of the use of LNG over NG, which itself is a very attractive engine fuel, is mainly that it is readily transportable, convenient to store, and since at the point of consumption the liquid gas must be vaporized, an enormous potential for refrigeration exists. It suffers, however, from the fact that a cryogenic tank is needed to maintain the -161°C temperature necessary to keep the fuel liquid, which is both costly and bulky. Moreover, there is an increased tendency for air pollution due to the emission of hydrocarbons as a result of the evaporation of the gas during periods of starting and inactivity.

Attempts were made in the United Kingdom before and during the war to use LNG for powering motor vehicles, (7). This work was later abandoned when there was no special strategic need for it. Currently various projects are going on, particularly in Northern America to demonstrate that LNG can be used as a fuel in mobile equipment under various conditions, (8). As part of a programme of LNG research at The University of Calgary, Canada, the combustion characteristics of LNG are being examined. Of special interest is the nature of the problems brought forward by using liquid methane in conventional combustion devices.

Thermal power applications involving the use of liquid natural gas, emply a cryogenic fuel tank where the liquid fuel is fed to the engine through a heat exchanger. The liquid is converted to a gaseous state and mixed with the necessary air, in a conventional carburettor or fuel injection systems. Apart from the problems associated with the handling of a cryogenic fuel, the most significant difference in the operation of engines on LNG rather than NG, is in the potential cooling possible of the charge or some selected engine parts. This is particularly attractive for high output turbo-charged piston engines where the refrigeration effect can be used to lower the overall intake charge temperature thus improving the power output and reducing the tendency towards knock and preignition. As shown in Figure 2, for a dual fuel engine running on methane, the knock free output is dependent exponentially on the inverse of the absolute intake temperature. Thus the potential improvement in output would be particularly substantial. However, light load operation of the dual fuel engine will be impaired somewhat by a cooler overall charge temperature due to the use of LNG.

With LNG fuelled spark ignition engines, much higher compression ratios, or boost pressures, become permissible, as shown in Figure 3; thus improving the specific output and efficiency. The refrigerating effect of LNG also maybe used indirectly towards improving the intercooling between the compressor stages of the turbo-charged engine or cooling the waterjacket temperature. This would be particularly attractive in mobile and traction applications where any reduction in size and weight of equipment is attractive. The slight cooling of the charge, however, would tend to increase the tendency towards cyclic irregularity and exhaust emissions.

There has been some interest recently in the future applications of LNG as a fuel in advanced aircrafts, (9). The most significant attraction is the relatively high heat sink of LNG which can be exploited to improve performance; in addition to increased engine hot section life and improved safety.

The application of LNG to the combustion chamber will have a substantial contribution towards reducing the heat transfer to the casing, improved regeneration and may be used to cool turbine blades thus enabling higher working temperatures and producing better efficiency and specific output. Some relative improvement in the compressor work may also be obtained. However, direct mixing of some LNG with the induction air prior to compression to lower the initial cycle temperature would be both operationally and thermodynamically unattractive.

CONCLUDING REMARKS

Many of the technical problems arising from the applications of LNG in engines are similar to those when NG is the main fuel. Much research and experience exist in this area. There is, however, very much less information regarding the further problems brought about by the relatively high heat sink of LNG when used as an engine fuel. Future expansion in this application of LNG is very much dependent on the further technological advances needed in the handling of cryogenic fluids for everyday applications. Selected replacements of conventional fuels in the transportation field will be favoured in applications where there is common facilities for handling and storage of the fuel and a high engine utilization factor. Therefore, its application in the private vehicle field is unlikely to be competative with conventional liquid fuels in the foreseeable future.

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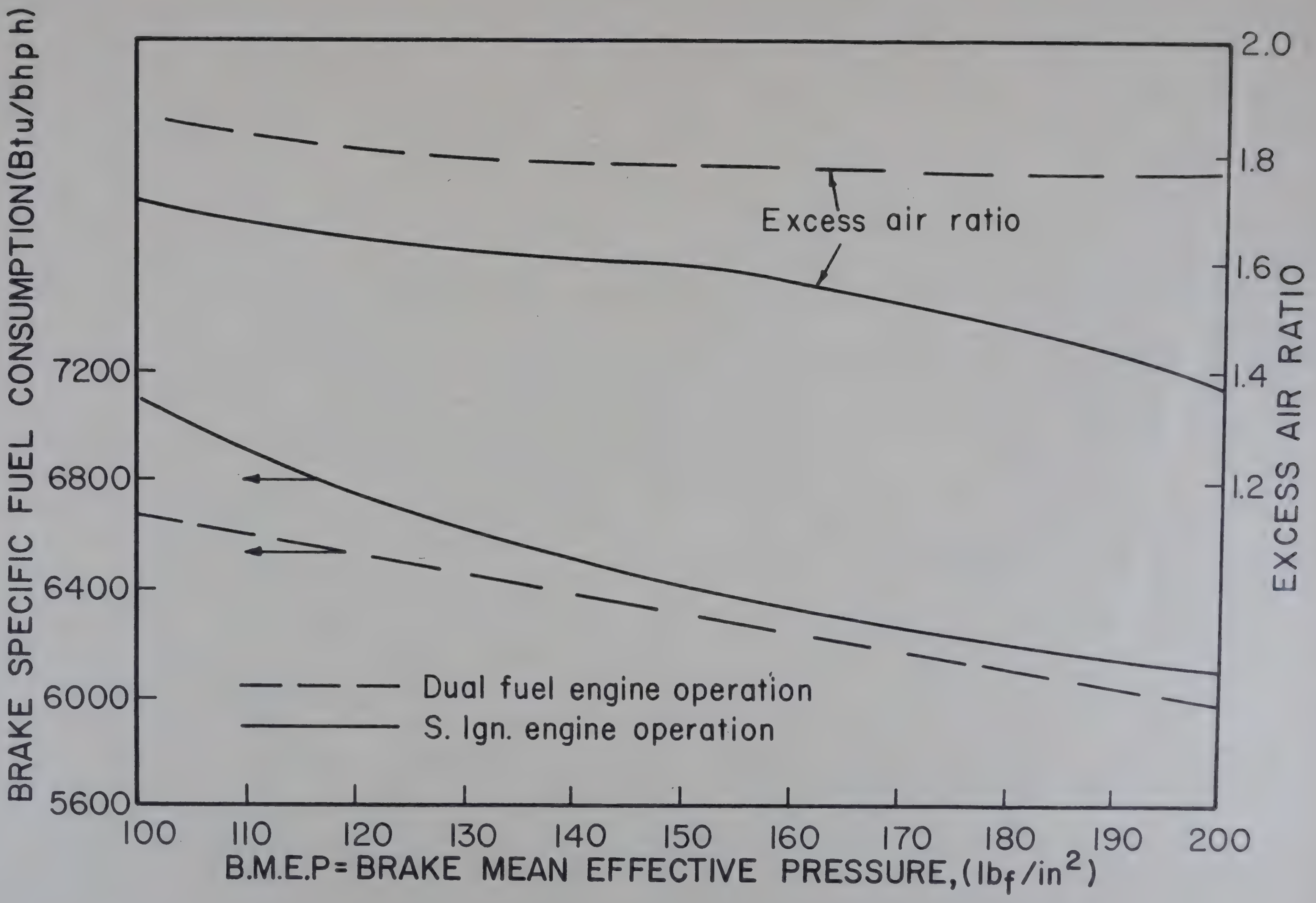

Fig.1. Typical performance curves for spark ignited and dual fuel versions of turbo-charged engines operating on natural gas, reported by M. Land (4).

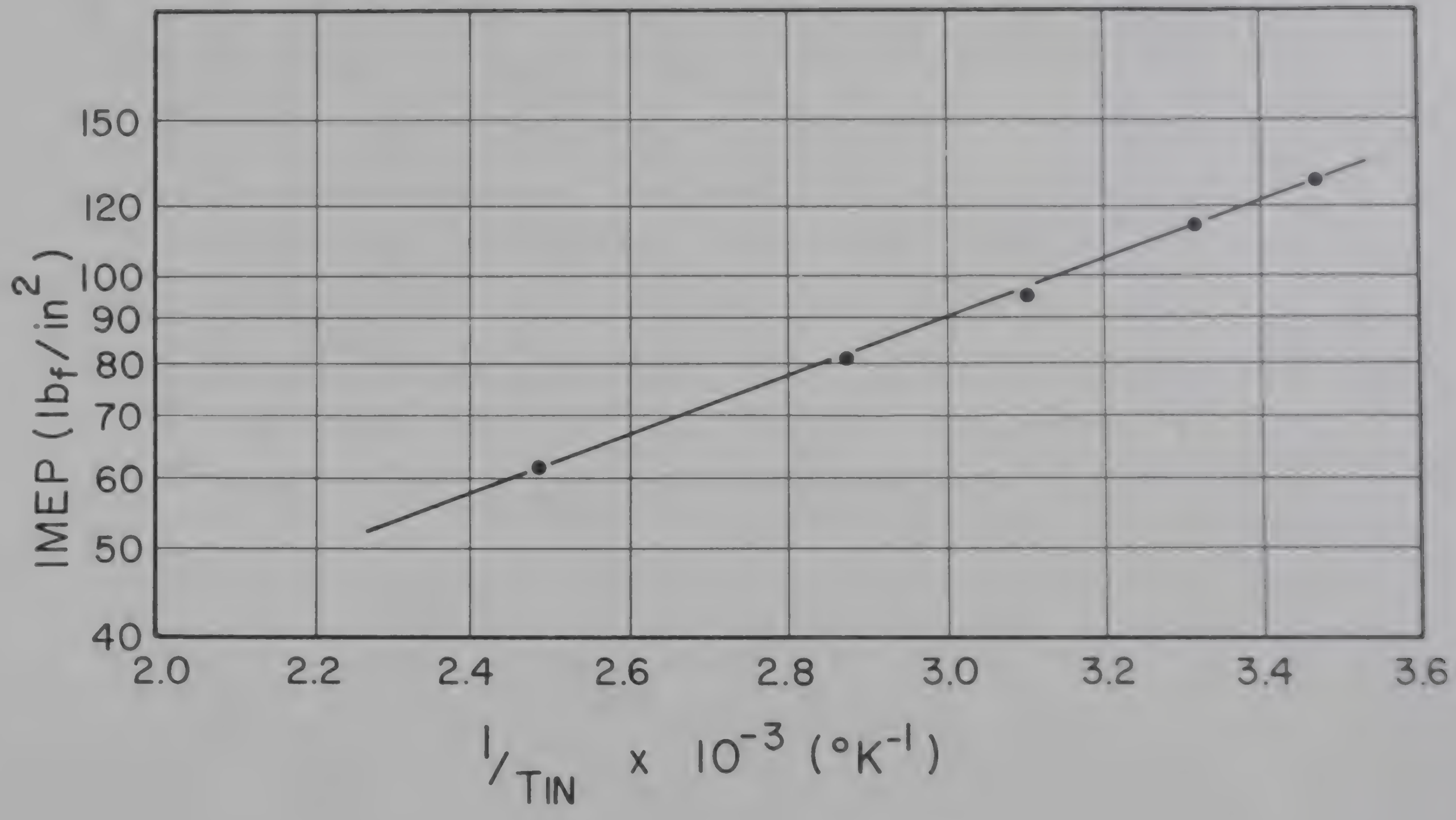

Fig.2. Correlation of the knock free output of a laboratory single cylinder Gardner engine operating as a dual duel on methane with the reciprocal of the absolute intake charge temperature in °K.

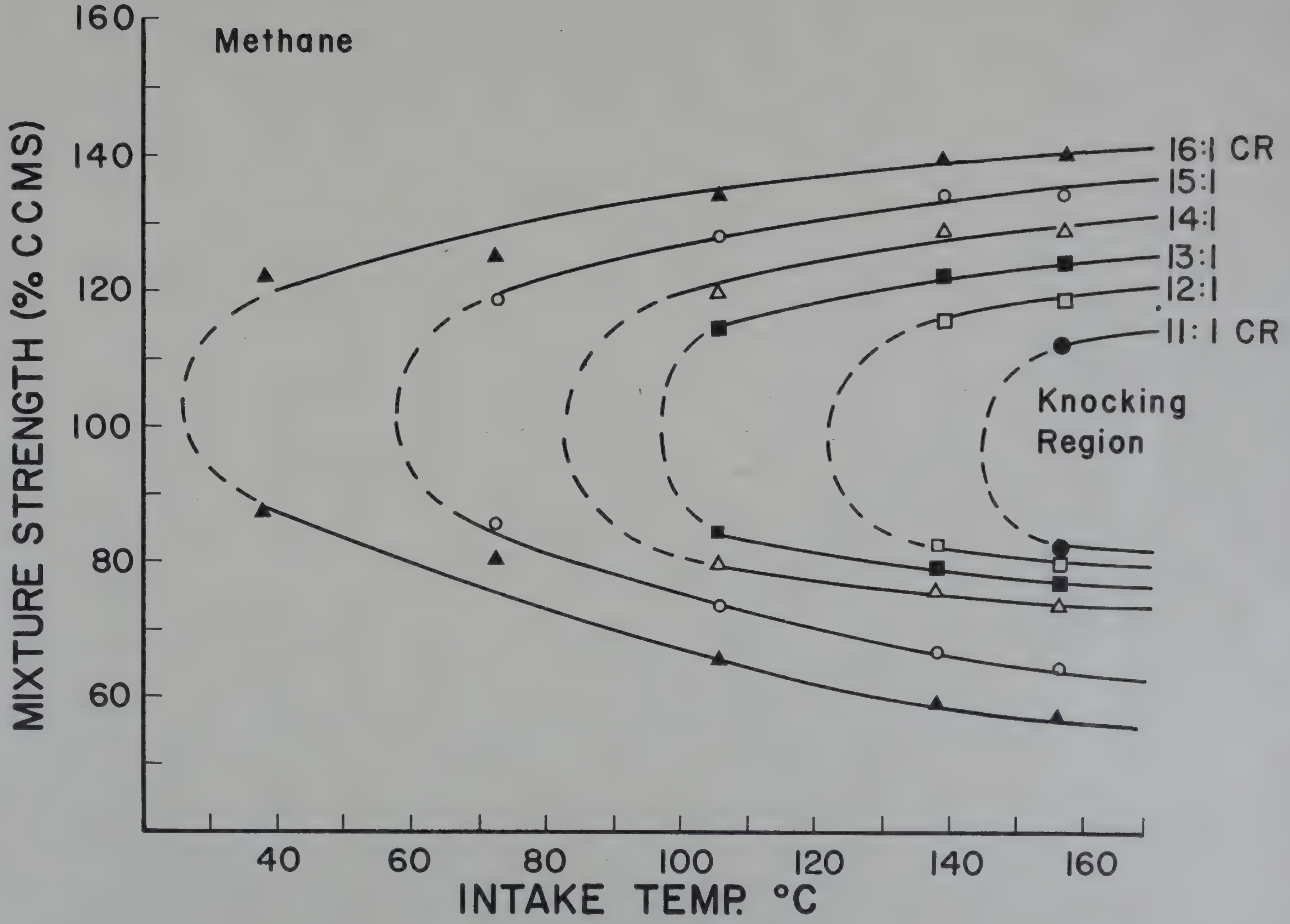

Fig.3. The variation of the knock limited compression ratio of a CFR single cylinder engine with methane as a fuel, with changes in the charge temperature (2).

GROUND REFUELING SYSTEMS FOR LIQUID-METHANE FUELED SUPERSONIC AIRCRAFT

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INTRODUCTION

Liquified natural gas (LNG) as a fuel for supersonic transport (SST) is mentioned in a majority of the published reports and trade journal articles on the subject of future applications of LNG. A recent extensive industry survey indicated that there is active interest in using LNG as an energy source for propulsion systems. Theoretical analysis by the National Aeronautics and Space Administration (NASA) in the United States has shown that a pay load improvement up to 31% and a direct operating cost reduction up to 36% might be realized by using methane rather than JP fuel in a supersonic aircraft. These figures have been the catalyst for current studies of improved engine components, tankage and fuel systems and insulation techniques which can be used efficiently with the cryogen.

This study assumes that the concept is technically feasible and that there are no fundamental barriers which prevent the solution to aircraft design problems, and is therefore only directed to an investigation of various methods for supplying liquid methane to the aircraft. The first task of the systems analysis was to define the design criteria for the ground refueling system, namely, what does the aircraft fuel system require in terms of quality, flow rate, temperature, and pressure; what volumes of fuel are required; and where is the fuel to be used. Various combinations of sub-systems are then analyzed to develop a support system which is most suitable for a specific application.

During the preparation of this report it became apparent that the cost of handling the fuel after it leaves the local airport storage must be given careful consideration when developing the price per BTU delivered to the aircraft. It is the purpose of this paper to introduce industry to the complexity of distribution and terminaling of a cryogenic aviation fuel.

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SYSTEMS REQUIREMENTS

Improved performance of a liquid methane fueled turbojet powered supersonic aircraft is derived from the greater heating value by weight and higher heat sink capacity of methane. The ground equipment, as well as the aircraft, must be designed to combat the two major disadvantages: a greater volume requirement; and the cryogenic nature of the liquid methane. Analytical studies have been made by the NASA of three example systems for aircraft tankage: (1) saturated methane loaded into nonintegral high-pressure tanks, (2) subcooled methane pressurized by a no-loss helium system, and (3) a combination of saturated liquid methane loaded into high-pressure tanks and subcooled liquid methane with methane gas pressurization by means of a stand pipe. Although ground systems for servicing each of these concepts have been investigated, the scope of this report is limited to a discussion of systems for the high pressure, non-integral tanks which require a saturated fuel.

The basic criteria for establishing a design flow rate for the refueling system are aircraft tank capacity and time available for fueling at a station stop. Using the specification for the Boeing 2707-200 as a basis for calculation and considering the additional volume required due to the lower density of methane, the methane fueled supersonic transport will have a fuel tank capacity of approximately 372 cubic meters (m^3). An analysis by the United States Federal Aviation Administration (FAA) of the SST design and size of ground facilities expected in the 1974-1990 time period indicates a 45 minute ground time for the SST. Applying other factors to these parameters, a flow rate of 0.15 cubic meters per second (m^3/s) seems to be practical. The subject of fuel quality has always received considerable attention by the airline industry. Since methane will not be excluded from such agenda, it is important that consideration be given to establishing a preliminary specification for an aviation grade liquefied natural gas. Although a misnomer, throughout the remainder of this report we shall refer to a product which is composed of the following constituents as LCH_4 or liquid methane wherein percentages are by weight: Methane greater than 89%; other hydrocarbons, such as ethane, propane and butane, less than 10%; nitrogen less than .2%; CO_2 and H_2O nil; sulfur less than .3%; solid particle contamination less than 1 milligram per 1000 cubic centimeters.

FUEL DEMAND

The projection of SST fuel requirements are based on a comprehensive review of industry and government market analyses which attempt to predict the sales volume for the U.S. SST through the year 1990. These projections consider the impact of a ruling that would prevent the SST from flying over populated areas. This restriction, commonly referred to as a sonic boom restriction, will reduce the number of routes that can be flown and ultimately the number of aircraft required for a given period. Airport congestion is a relatively new problem which is currently being investigated and could have an impact on the total quantity of fuel consumed at high density airports, such as New York and Los Angeles.

The annual world-wide and daily local fuel consumption quantities have been developed from the amount of fuel consumed by one aircraft on an average trip and the number of trips each aircraft can make in one day. In Figure 1, the black solid line is the theoretical block JP fuel consumption for the Boeing Model 2707-100. (Block fuel consumption refers to total fuel used per trip for taxi, take-off, climb, cruise, desend, land and taxi at the destination.) The dashed line displays the additional 80% methane required to fly an equivalent length trip. For the purposes of this study we have used an average trip length of 4,022 kilometers (km). Therefore, if the distance between the origin and destination is 4,022 km, 101,600 m^3 of gaseous methane, herein termed CH_4, will be consumed on an average trip. Aircraft utilization is expected to increase to slightly less than 9 hours per day during the 1980-1990 time period. Cruising at a speed of Mach 2.7, the block time to fly the average trip is 2.10 hours. One may then expect each aircraft to make in excess of four trips per day.

These hypotheses have been used to generate Figure 2 which relates the available aircraft to the annual CH_4 demand. In the 1990 time frame with no sonic boom restriction, world-wide demand for aircraft could be as high as 1170, but if the SST can not operate over populated areas because of boom restrictions the number of revenue passenger miles that can be generated by SST's will be reduced considerably but will yield a demand for approximately 500 aircraft. The uncertainties inherent in long range forecasting favor the conservative approach of assuming this latter figure, in which case the entire SST fleet could consume approximately 79,000 million cubic meters (79 Gm^3) of gaseous methane per year. This does not include fuel for liquefaction and losses sustained during transportation.

The John F. Kennedy (JFK) International Airport in New York City is an interesting example of the fuel demand at one facility.

BLOCK FUEL CONSUMPTION U.S. SST

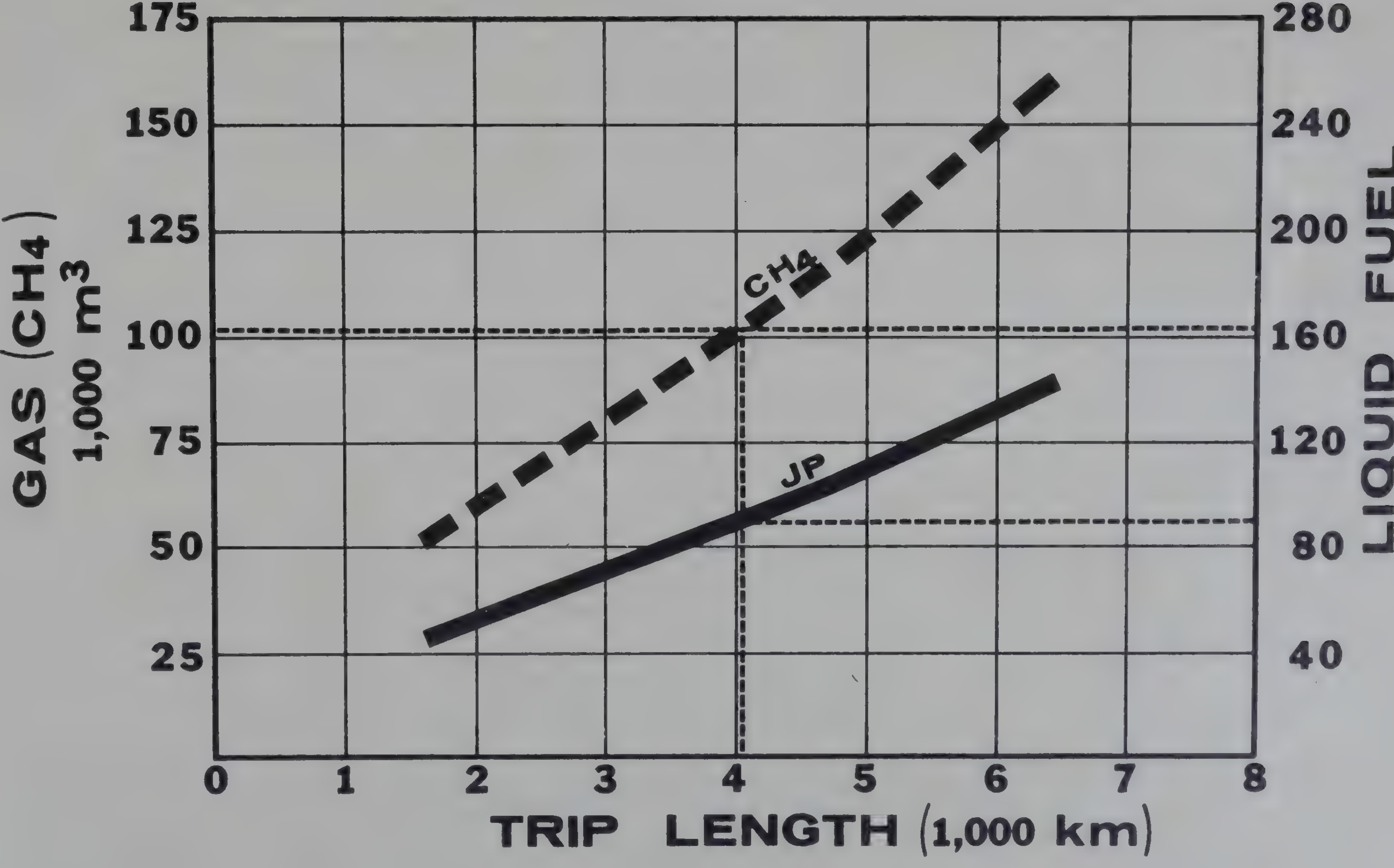

Figure 1

U.S. SST FUEL REQUIRED WORLD WIDE

YEAR	NUMBER OF AIRCRAFT	ANNUAL CH_4 CONSUMPTION G(m^3)
	NO SONIC BOOM RESTRICTION	
1980	485(3)	77
1990	1170(3)	186
2000	1750	278
	WITH SONIC BOOM RESTRICTION	
1980	207(3)	33
1990	497(3)	79
2000	750	119

Figure 2

Using the Boeing Company Airport Activity Analysis for 1976 and 1980 as a baseline, the number of daily departures for JFK have been extrapolated through the year 2000 (see Figure 3). Applying the average block fuel consumption developed previously, we conservatively estimate a daily requirement of approximately 60,000 cubic meters (60 k(m^3)) of liquid methane or 38 million cubic meters (38Mm^3) gaseous methane in 1990. Figure 4 illustrates the volume of liquid methane to be serviced at specific periods during the day. The irregularity observed here will have a major impact on the design of the airport fuel system.

GROUND REFUELING SYSTEMS

With a general understanding of the aircraft interface requirements and the projected quantities of fuel to be consumed, we are ready to examine and assemble the elements of a total ground refueling system. For this analysis we are interested in all of the major functions involved from gathering of natural gas at its source to the onloading of liquid methane at the aircraft refueling receptacle.

In Figure 5 we have classified the alternate basic supply subsystems by the geographical location of the liquefaction plant relative to the airport distribution system. In the first case, natural gas from domestic reserves is moved through major transmission lines to a liquefaction facility located in the vicinity of the aircraft storage tanks. In the New York City case, a pipeline more than 100 centimeters in diameter would be required from the source of gas reserves to supply the daily feed gas for the liquefaction plant.

In the second case, liquefaction in the field near the source of natural gas will require overland transportation of liquid methane in the form of rail car, motor vehicle, tankers or pipe line. If we apply the unit train concept to the New York example, it would require 464 rail cars, each having a capacity of 129.46 m^3, to supply the 60,000 m^3 daily consumption.

In a third situation where a group of landlocked, lower traffic density airports are to be supplied with liquid methane, we may consider using the remote liquefaction approach. In the Midwest of the United States, SST activity is forecasted for St. Louis, Kansas City, Minneapolis, St. Paul, Omaha and Sioux Falls. The limited quantity of methane to be consumed at each of these airports favor the installation of a liquefaction plant on a major gas transmission line at a central location, such as Davenport, Iowa.

NEW YORK JFK

YEAR		DAILY DEPARTURES	DAILY FUEL REQUIREMENT LCH$_4$ k(m^3)	DAILY FUEL REQUIREMENT CH$_4$ M(m^3)
1976	A	87		
	B	56		
1980	A	241	39	24
	B	154	25	16
1990	A	580	94	59
	B	**371**	**60**	**38**
2000	A	870	141	88
	B	556	90	56

A: NO SONIC BOOM RESTRICTION

B: WITH SONIC BOOM RESTRICTION

Figure 3

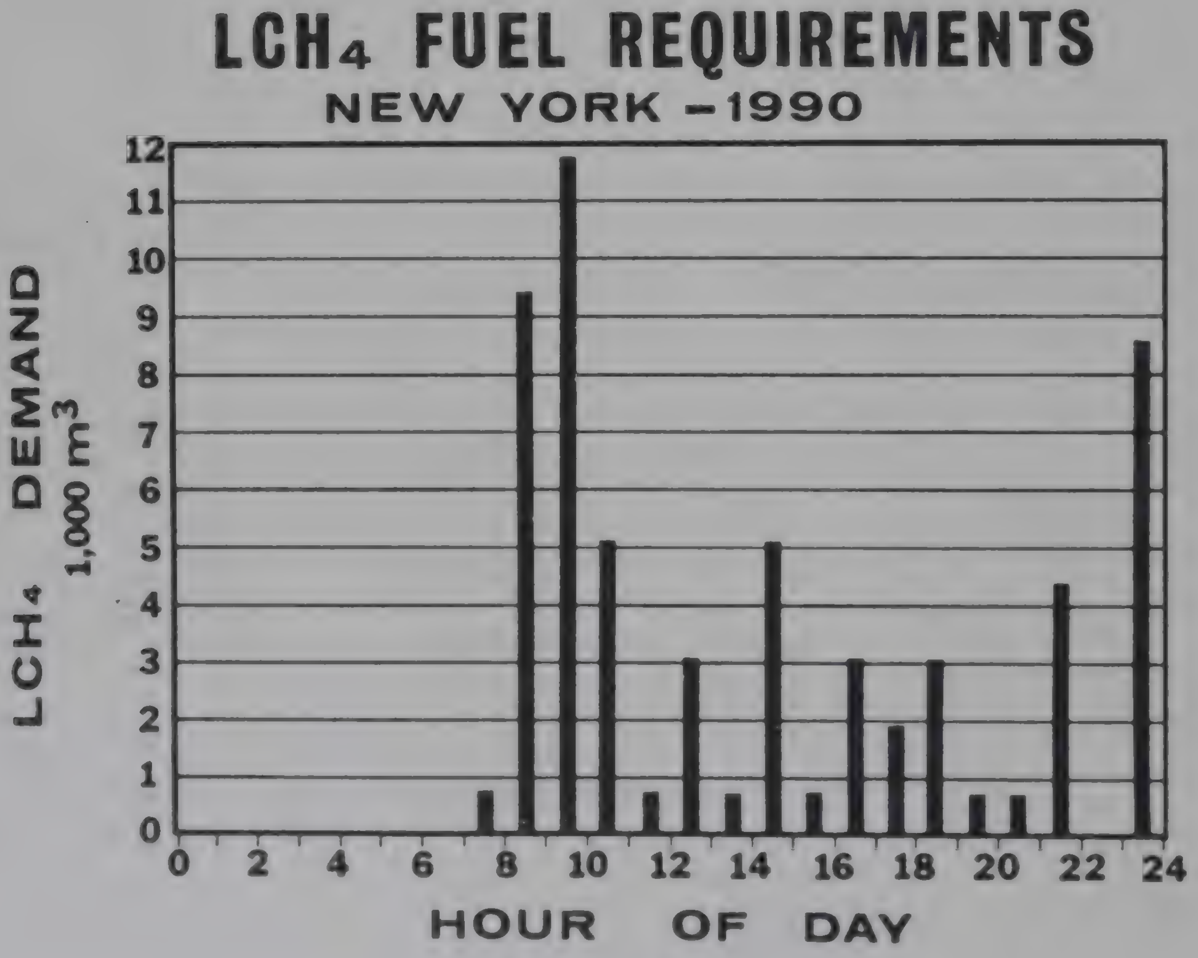

Figure 4

SYSTEMS DIAGRAM

NATURAL GAS TO LIQUID METHANE AT THE AIRCRAFT

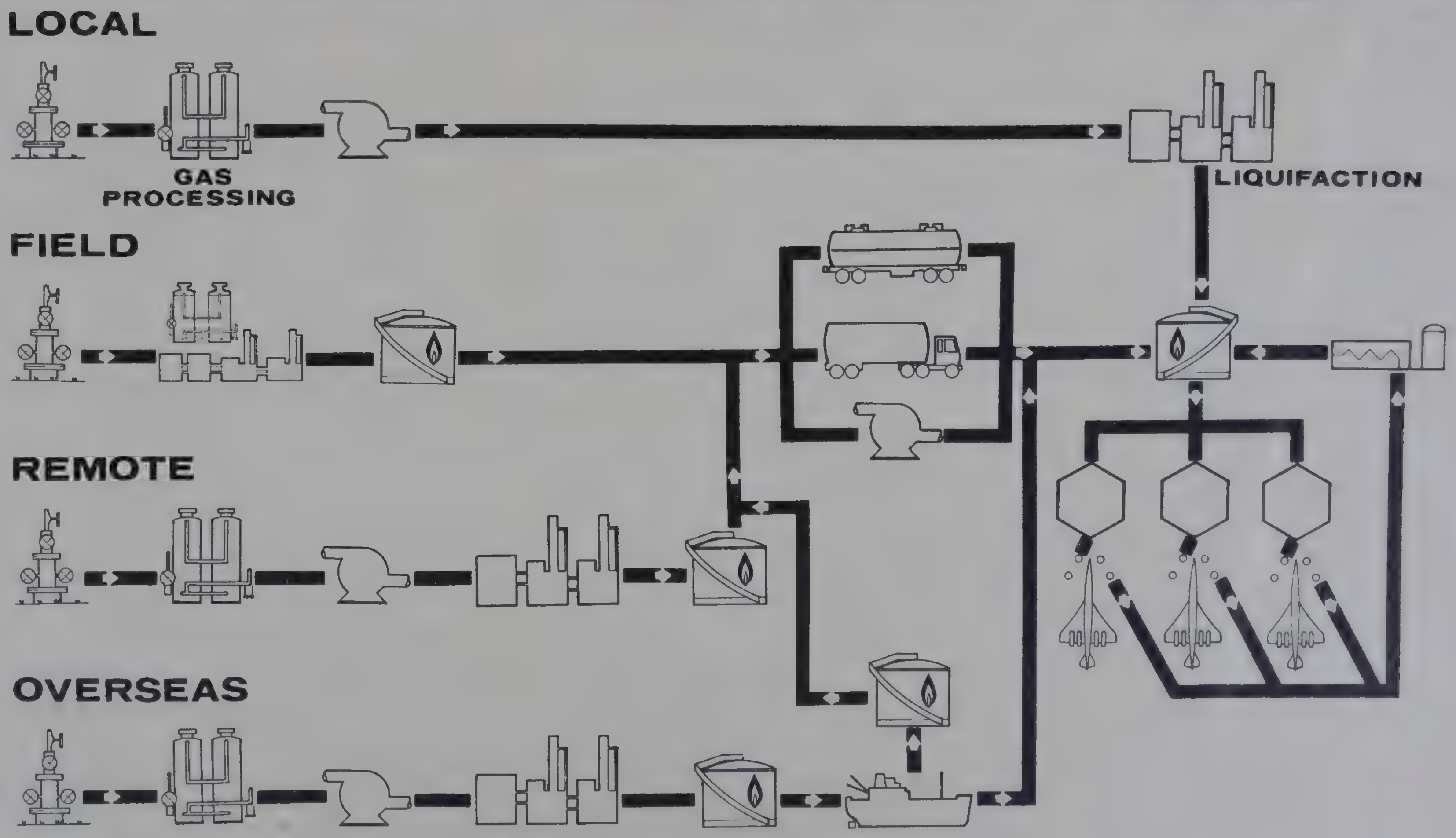
LOCAL
GAS PROCESSING
LIQUIFACTION
FIELD
REMOTE
OVERSEAS

Figure 5

In this case, we would also investigate the potential for barge transportation along with the other modes previously described.

The fourth scheme takes advantage of foreign gas reserves and provides an economically attractive method for delivering large quantities of liquid methane to airports in the vicinity of seaports. In countries where the domestic reserves are inadequate, ocean tankers in conjunction with overland transportation systems will be mandatory for supplying the landlocked installations. Using tankers the size of the Methane Princess for comparison, it would take 2-1/2 tanker loads per day to supply the daily fuel to be used at JFK in 1990.

Choice of a basic supply system is influenced by the same factors that we would consider in the design of a base utility system, namely total demand, source of raw material and location of the consumer. Continuing with the system's diagram, we will move directly to the terminal distribution sub-system in order to prequalify some of the major elements of the airport storage and processing sub-system.

A better understanding of the requirements of this system may be accomplished by review of some of the ramp equipment and procedures for refueling a liquid methane fuel supersonic aircraft which uses the pressurized, non-integral tank concept. For this discussion, refer to Figure 6. A majority of the published plans for terminal facilities designed to accomodate the U.S. SST, favor straight end or diagonal parking at the gate position. In either case, four fuel service connection assemblies will be located in an underground storage compartment directly beneath the aircraft refueling receptacle. These compartments are illustrated by the four black rectangles on the left side of the diagram. A simplified flow schematic of the typical gate position is shown in the exploded view to the right side of the diagram. Eight liquid service lines are considered a minimum to attain the 0.15 m^3/s refueling rate previously defined. In addition, four vapor return lines must be provided to accomodate the boil-off vapor generated during tank chilldown and hold.

An articulated powered swing arm will be used to automatically elevate and support the three flexible lines and quick disconnect panel assembly. The distribution system is vastly more complex than is illustrated by these simplified diagrams. For example, a preliminary layout of the quick disconnect panel assembly indicates that it will consist of: three self-sealing hydraulic quick disconnects equipped with pneumatically operated shutoff valves

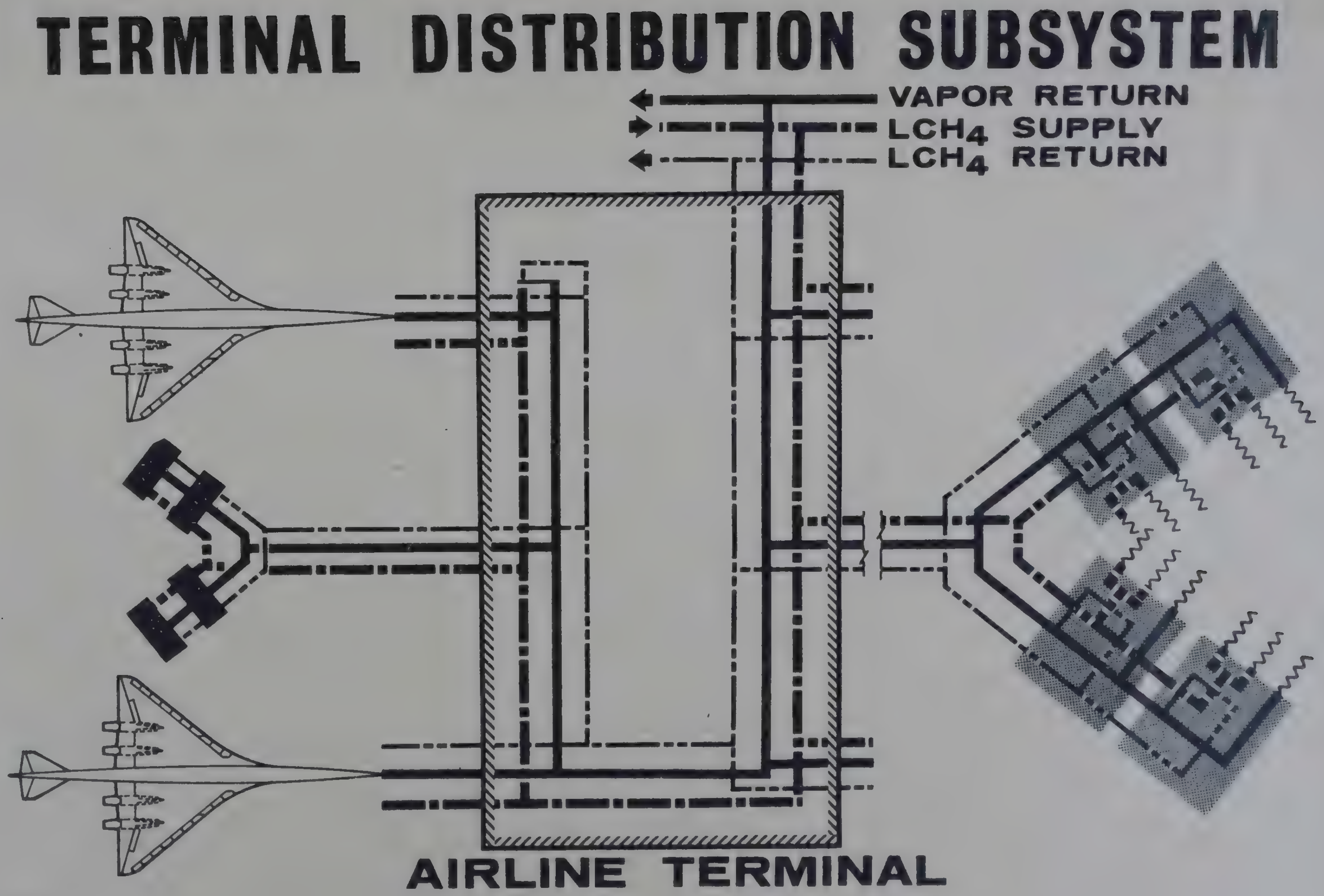
TERMINAL DISTRIBUTION SUBSYSTEM
VAPOR RETURN
LCH$_4$ SUPPLY
LCH$_4$ RETURN
AIRLINE TERMINAL

Figure 6

and manual overrides; three pressure transducers, three temperature transducers, one electrical quick disconnect; one static ground connection; a pneumatically operated locking mechanism, and a mechanically operated disconnect sensing device.

Onloading of a saturated liquid into the aircraft tanks is more efficiently accomplished by continuous circulation of the fluid through the liquid supply manifolds; in essence this treats the supply manifold as an extension of the storage tanks. Otherwise, transmission line heat gain must be offset by subcooling at the origin of the distribution system. Fluid circulation is maintained as close to the aircraft interface as possible as shown by the connection of the liquid supply line to the liquid return line in each underground compartment. A valve arrangement is incorporated to direct the fluid to the liquid return line during offloading (or aircraft defueling) of large quantities of liquid methane. The bypass from the liquid supply to vapor return is activated for distribution system line chilldown. A similar bypass configuration to those just described is used in the terminal manifold for initial facility start-up, maintenance and reconditioning after emergency shutdown.

Within each of the underground compartments, it is planned to incorporate the origin of the boil-off management scheme. The non-integral tanks may not be subjected to internal pressures in excess of 2.06 kilograms per square centimeter (2.06 kg/cm^2). Considering safety factors, a back pressure control unit will be designed to sense and relieve positive pressure in the vapor return line in excess of 350 grams per square centimeter (350 g/cm^2) above ambient. Fail safe devices will be included to prevent inadvertent exposure of the aircraft tanks to a negative pressure.

The purpose for some of the components in this sub-system can be described by a review of the sequence of events relative to the refueling operation during the 45 minute ground service period. Immediately after the engines have been shut down, the automatic swing arms will extend from the underground storage compartment and lock onto the aircraft refueling receptacles. Positive aircraft tank pressure in excess of 1.03 kg/cm^2 which was generated by boil-off during the flight is relieved through the vapor return line. The relatively small amount of fuel remaining in the depleated tanks will have vaporized during the flight and will not be adequate to maintain the tanks in a cold condition. Therefore, the tanks which contained the block fuel consumed by the previous flight will be empty and "warm", thus requiring chill-down and cold-soak.

Since detailed designs of aircraft tanks for storing liquid methane were not available at the time this paper was prepared, we have used information from preliminary studies by the U.S. National Aeronautics and Space Administration, as reported in Reference No. 2, to estimate the time and volume of liquid fuel required to condition a hypothetical tankage configuration. For ground equipment design purposes, we assume the extreme case in which all of the tanks are empty and warm. For this condition we should expect to consume 3.03 m^3 liquid methane during a 142 second chill-down and 0.92 m^3 for cold soak during a 10 minute period after liquid fill is complete. This includes conditioning of the flexible lines. On less than maximum range flights, a predetermined quantity will be loaded in selected tanks. In a majority of cases, the reserve fuel will not have been used during the trip. By venting the reserve tank at the start of the refueling operation, it should reach equilibrium and be ready for topping off by the time the other tanks are filled. During the semi-automatic fill operation, the following parameters will be monitored automatically or manually: rate of flow; quantity; fuel inlet temperature and pressure; vent line pressure; tank temperature and pressure; and tank liquid level. The vapor return line will remain hooked up to the aircraft until just prior to engine start and roll out to prevent premature tank pressurization.

The preceding discussion has been directed to one airline terminal and one gate position. We will now examine the subsystem which must store and process the total volume for an entire airport. Although Figure 7 illustrates only four terminals, one should recognize that there could be as many as eight or ten branches off of the main supply manifold loop. The concept presented for this sub-system is one of the least complex versions of the group that we have developed and analyzed in recent months. Time does not permit a description of methods for servicing the helium pressurized, subcooled tanks or combination tankage concepts. It is also unfortunate that we cannot cover the many variations to this system. As an example, a central reliquefaction facility for processing the boil-off gas from the distribution subsystems as has been illustrated. This operation could be divided into smaller units located at each terminal or possibly in each underground compartment on the apron. These trade-offs are subjects to be considered in the detail design of a specific installation.

A key component of the liquid supply manifold circulation loop is the reconditioning unit identified as a sub-cooler in the

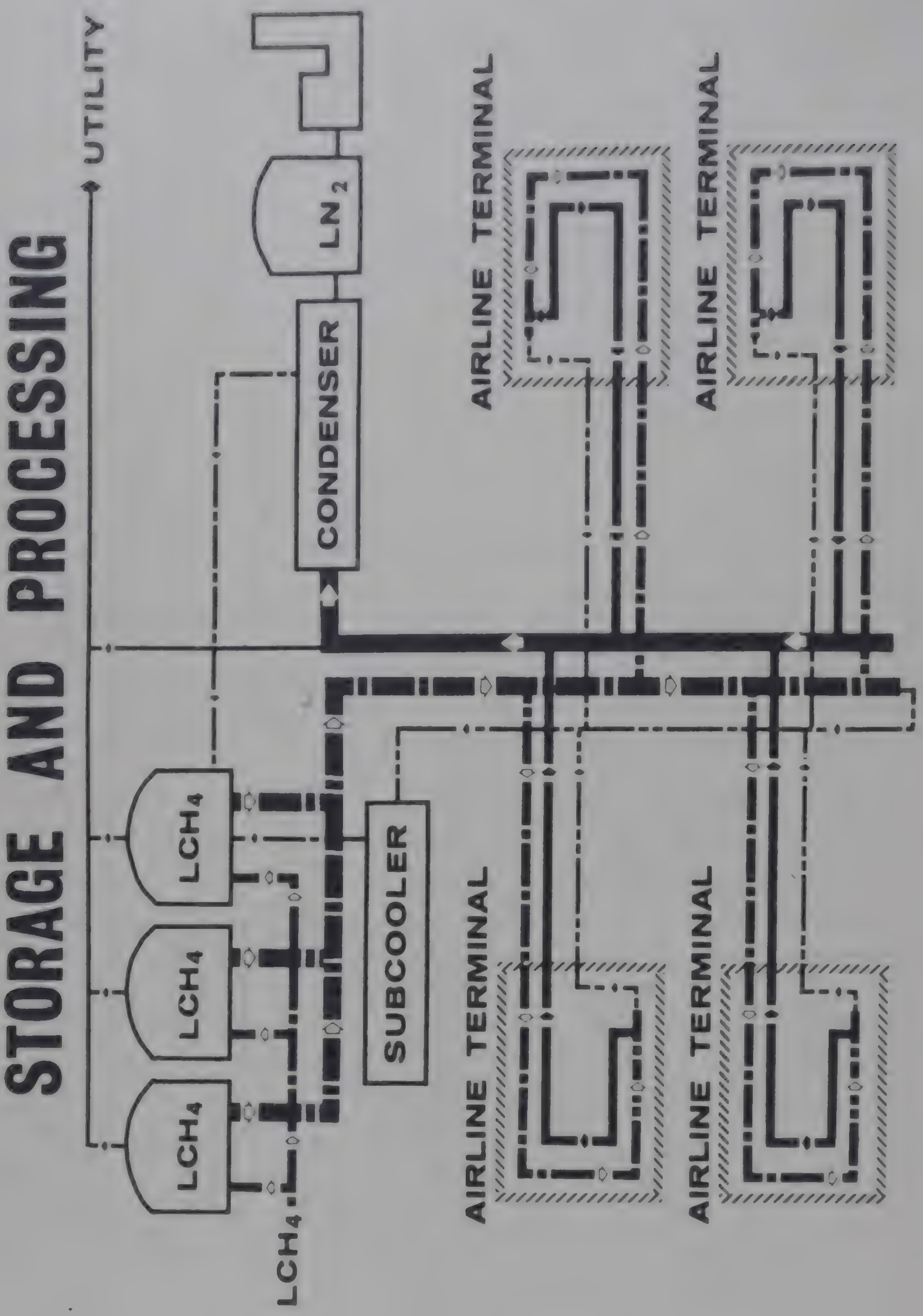
STORAGE AND PROCESSING
UTILITY
LN$_2$
CONDENSER
LCH$_4$
LCH$_4$
LCH$_4$
LCH$_4$
SUBCOOLER
AIRLINE TERMINAL
AIRLINE TERMINAL
AIRLINE TERMINAL
AIRLINE TERMINAL

Figure 7

diagram. The rate of fluid movement in the liquid supply manifold will be predetermined to assure that saturated liquid reaches the most distant point in the supply line. In the liquid return line we expect to experience a heat gain which will be a function of line length and design economy. The sub-cooler will be sized to reduce the liquid temperature to 112° K or less prior to entering the storage system.

It is interesting to compare the size of today's storage and transfer facilities with one which would be used to store and process the quantity of fuel required in the New York example. Four tanks similar to the 56.6 million cubic meter (56.6 M(m^3)) (gas) prestressed concrete tank presently being constructed on Statten Island, New York for the Texas Eastern Transmission Corporation would be recommended to insure uninterrupted service of the 38 M (m^3) (gas) daily aircraft fuel consumption expected by 1990. Allowing for non-recovered loss due to boil-off and handling, the 226 M (m^3) of gas storage would provide one day's supply with a five day back-up. This reserve is considered desirable particularly when ocean transportation is involved. The maximum liquid fuel demand for a one hour period as shown in Figure 4 is 11,700 m^3 from 9 to 10 a.m. This was derived from a prognosis that 72 aircraft will depart New York for various destinations during that period. Assuming that 60% of these aircraft will be refueling at one time, 121.9 cm line would be estimated to accommodate the 6.51 m^3/s flow rate in the main liquid supply manifold.

The vapor return line is designed using the similar criteria except the time of tank chill-down is much less than that consumed on loading the liquid. A reasonable assumption for the maximum number of aircraft generating chill-down boil-off at one time would be in the order of 5% of the 72 aircraft. This line must also accommodate vapor displaced from aircraft tanks receiving liquid as well as the boil-off generated during hold prior to take-off. Operating at a line pressure of 3.51 kg/cm^2 at the condenser, a 76.2 cm line is estimated for this service. It is obvious that it is not practical to process all of this peak vapor load through the condenser system. Sale of the surplus gas to a large utility appears to be an attractive approach to improving the overall economy of the system operation.

CONCLUSION

Analysis of the three major subsystems has been presented to illustrate the inner relationship of specific elements and the effect of system for supplying liquid methane to a supersonic aircraft. The major dissimilarities between these concepts and those techniques used to refuel subsonic aircraft are size and system complexity. The magnitude of these differences will prompt a change in the current philosophy of facility planning, ownership and operation.

Historically, the development and acquisition of a full complement of ground support equipment has lagged the design, manufacture and inauguration of scheduled service for a new fleet of aircraft. The similarity between preceding generations of aircraft has permitted this inefficient and uneconomical practice to survive. However, the cryogenic nature of liquid methane makes it impractical to modify, enlarge or upgrade existing ground refueling facilities. The lead time to activate new liquefaction plants; commission a fleet of ocean tankers; erect storage facilities; and install a new airport terminaling system invokes the requirement for a concurrent aircraft and ground equipment development program. Otherwise, the success of this new concept would be jeopardized not by a shortage of, but rather, by non-existent support facilities.

The potential impact on ownership and operation deserves consideration. A number of the international airports in the United States are served by more than one fuel supplier. Each of these companies owns and operates a separate system for distributing a particular brand of fuel to preferential customers. The inefficiency and resultant increased cost of operating duplicate cryogenic systems will preclude this practice. Multiplicity of heat gain sources with the attendant requirement for additional reliquefaction equipment is only one example of the penalties of parallel systems. Project cost is another factor which will influence business policy. Additional indepth studies will be necessary to accurately evaluate the cost of a total system for supplying liquid methane to various points of utility. However, it is anticipated the tremendous quantity of methane required to support this application will be a strong incentive for industry to participate in joint ventures. To illustrate this point, the forecasted daily base load of the JF Kennedy airport in New York is approximately twice the combined output of all the operational liquefaction facilities in the world today.

Liquefied natural gas as an energy source for advanced propulsion systems if only one of many interesting potential new uses for LNG. Effective exploitation of this rapidly expanding market will be accomplished by progressive organizations that actively participate in developing advanced technology for processing, transportation and handling of this cryogenic energy.

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170	Arbella, R.	Director, Rooo-Verken, Suenungsson & Co, Sweden.
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